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Translation

Absorption of Vibration on Ships

By

A. S. Nikiforov

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ABSORPTION OF VIBRATION ON SHIPS

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ANNOTATION

[Text] This book examines various aspects of the problem of reducing sonic vibration which is set up in ship structures during equipment operation, causing increased air noise in compartments of the ship. It describes the working principle and design of vibration absorbing installations. It makes recommendations on the rational use of means of vibration absorption on ships and gives methods for evaluating their acoustical effectiveness. The book describes technology for manufacture of some vibroabsorptive materials.

This book is intended for technician-engineers and scientific workers who are involved with the questions of reducing sonic vibration and air noise on ships. It may be of interest to specialists working with the problems of reducing the level of sonic vibration and noise in automobiles, trains and aircraft. The book will be useful to students and post-graduates specializing in the aforementioned field of acoustics.

Reviewed by Professor I.I. Klyukin,
Doctor of Technical Sciences

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PREFACE

Growth in the power and resultant vibration of ship machinery carries with it increased sonic vibrations in the ship's hull-frame structures and a subsequent increase in the noise levels inside compartments of the ship. Development of measures oriented toward reducing vibration takes two primary directions -- insulation of vibration and absorption of vibration. The former has been covered in detail in literature [5, 24]; the latter, despite the high effectiveness of means of vibration absorption, has not to the present been systematically examined.

This book generalizes the experience of applying means of absorption vibration on ships and other forms of transport, as well as in industry. In so doing it uses domestic and foreign publications and the author's own experience in this field. However, not all aspects of the vibration absorption problem are examined here in equal detail. For example, questions of the chemical structure of vibroabsorptive materials and its relationship to physical and mechanical properties have been ommitted. For information on these questions the work [53], in particular, is recommended to the reader. Methods for experimental study of the dissipative properties of means of vibration absorption, which have been adequately described by S.V. Budrin in work [34], are not presented.

The reader may easily find information on general theoretical questions, which may be needed in reading of this book, for example, in monographs [18, 24 and 34].

We request that wishes and comments on the book be sent to this address: 191065, Leningrad, Ulitsa Gogolya, Izdatel'stvo "Sudostroyeniye".

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ARBITRARY NOTATIONS

B - cylindrical (flexural) rigidity
c - velocity of wave propagation
D - longitudinal rigidity
E - Young's modulus of material
f - frequency
G - shear modulus of material
h - plate thickness
I, Ip - axial and polar moment of inertia of cross section
$j=\sqrt{-1}$ - imaginary unit
k - wave number
M - mass of plate or rod
m - mass of unit of plate area (unit of rod length)
p - sonic pressure
q - energy flow
R - radius
S - area of plate
T - temperature, period of oscillation
t - time
w - density of energy
x, y, z - Cartesian coordinates
z - mechanical resistance
Э - effectiveness
Δ - difference of values
ζ $\dot{\zeta}$ $\ddot{\zeta}$ - longitudinal oscillatory displacement, velocity, acceleration of plate or rod sections
η - loss factor
θ $\dot{\theta}$ $\ddot{\theta}$ - deflection angle, angular velocity, angular acceleration of plate or rod sections
λ - wavelength
μ - ratio of values
ξ $\dot{\xi}$ $\ddot{\xi}$ - lateral oscillatory displacement, velocity, acceleration of plate or rod sections
ρ - density of material
σ - Poisson's ratio of material
φ - angle of incidence of wave
ω - circular frequency
v' - coordinate derivative (except as specifically noted)
<>v - averaging on parameter v
$\dot{v}$ - time derivative

Indices:

пл - plate
ст - rod
р - rigidity rib
п - longitudinal wave
и - flexural wave
с - transverse wave
к - torsional wave
вч - high frequency
нч - low frequency
вп - vibroabsorptive coating
ар - antiresonant
пот - potential
пог - absorbed
изл - radiated
и - and
кр - critical
эл - ellipse
н - upper
в - lower
вн - internal
с - shear
сж - compression
опт - optimum

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INTRODUCTION

Coulomb's work "Memoir on Torsion" (1794) must be considered the first study in the field of absorption of energy in vibrations of various structures and systems. Interest in absorption of vibration grew significantly by the mid-twentieth century. From 1920 through 1965 the number of papers on this subject increased from 17 to 2,500, exceeding the average growth factor for scientific-technical information [84].

Absorption of vibration is used in combating such adverse phenomena as sonic vibrations arising in and propagating through engineering structures, resonant oscillations of structure and system elements, auto-oscillation processes such as fatigue damage to materials, heating of parts under periodic deformation, etc. From the point of view of combating air noise in the compartments of ships and other forms of transport, we shall concern ourselves with the use of means of vibration absorption to reduce sonic vibrations and resonant oscillations.

Operating machinery, screw propellors and ship system accessories generate intense sonic vibrations in a ship's hull-frame which spread through it and cause air noise in compartments, sometimes very distant from the vibration sources. Under the effects of vibration of the hull, caused by the ship's travel, resonant oscillations can be generated in separate elements of the hull-frame, often leading to unpleasant jarring. Increased noise levels arise when the ship's bunkers are loaded with dry cargoes and coal, when anchor chains are hauled up or played out, when the ship is traveling in ice or in other cases which involve hard objects impinging on elements of the ship's structures. In all the listed cases vibration of the ship's structures and the resultant noise could be significantly reduced by the use of means of vibration absorption.

Two basic means of vibration absorption are most widely used in ships: vibroabsorptive coatings and special vibroabsorptive construction materials. The former are applied to finished structures to increase energy losses within them under periodic deformation. At present rigid, stiffened, pliable and other types of coatings are used. Structures possessing high dissipative properties, even without special coatings, can be produced from vibroabsorptive construction materials. Vibroabsorptive construction materials include laminated vibroabsorptive materials, vibroabsorptive alloys and glass-plastic.

Vibroabsorptive coatings and, to some degree, vibroabsorptive construction materials have the advantage over other means of soundproofing that they can be used on an already finished ship, in addition to the designed soundproofing system. The necessity for this might arise in case sanitary standards for habitation are not satisfied during ship construction or repairs. At the same time it should be remembered that means of vibration absorption which are provided for in design of the soundproofing

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system are less expensive than installation of the means on a completed ship.

Means of vibration absorption may be installed on structures to be damped both in a stationary fashion (for example, spraying on of a vibroabsorptive coating) and with the use of mechanical fasteners which allow removal or replacement if required. One should point out the use of such removable vibration absorbing installations, aside from their primary purpose, for damping of metallic structures during machining which generates intense vibrations and noise.

It goes without saying that means of vibration absorption will not totally eliminate vibration and noise in ship compartments. Complete resolution of this problem can be achieved only by the use of these means in combination with other soundproofing measures. At the same time, as will be shown below, the reduction by means of vibration absorption of vibration and air noise can be very great, reaching approximately 10-20 db. In some cases this permits the necessary reduction of vibration and noise to be attained by the use of only one means of vibration absorption.

The use of means of vibration absorption can lengthen significantly the useful life of vibrating machines and equipment, in certain parts of which fatigue damage is caused by prolonged and intense vibration. Work [78] shows that damping the blades of an air siren extended its operational life from 10 hours to 6 months.

Vibroabsorptive coatings and construction materials are widely used to combat vibration and noise not only in shipbuilding. Vibroabsorptive coatings are applied for this purpose to railway cars, locomotives, automobiles, tractors, aircraft, gas and oil pipelines, railroad rails and bridges, etc. Air conduits, sound-insulating housing and safety shields for machine chassis, charging hoppers, tracks for vibrating conveyors, etc. are made from vibroabsorptive construction materials.

Many researchers are at work in the field of creating vibroabsorptive coatings and materials and their practical use. In 1947 I.I. Klyukin suggested the design of a pliable vibroabsorptive coating for damping of ship bulkheads [20]. The theory of this coating was later developed by L.Ya. Gutin and foreign acoustics experts E. Ungar and E. Kerwin.

The first publications on rigid vibroabsorptive coatings were made in 1951 by I. Slavik and I. Nemets as well and P. Lienard. Concurrent with these works, G. Oberst, who made a great contribution in this field, began his own work on creation of such a coating. In our own country a group of specialists, led by B.D. Tartakovskiy, has been working on development of structures and materials for rigid vibro-absorptive coatings since the beginning of the 1950's. He is also credited with widely introducing means of vibration absorption into various spheres of the national economy.

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Stiffened vibroabsorptive coatings of structures was investigated by the American E. Kerwin in 1959 [77]. In the USSR the works of V.V. Tyutekin [16] on this question are well known.

Reference was first made to laminated vibroabsorptive materials (of the "sandwich" type construction) in 1956 by E. Stüber. Domestic materials of this type were created by B.D. Tartakovskiy, A.G. Pozamontir and their colleagues. A large contribution to the development and use of means of vibration absorption has also been made by B.D. Vinogradov, N.I. Naumkin, A.S. Nikiforov, M.I. Paley, L.I. Trepelkova and others. Among foreign specialists in this field mention should be made of D. Jones, D. Zboralski, G. Kurtze, E. Ungar and A. Schommer.

Through the efforts of domestic and foreign researchers significant experience has been accumulated in the development and use of means of means of vibration absorption. The most interesting information from this experience, important for shipbuilding, is commended to the reader's attention in this book.

Chapter 1. PHYSICAL BASES FOR ABSORPTION OF VIBRATION

§1. Absorption of Vibratory Energy in Oscillating Systems with Concentrated Parameters

The fundamental phenomena stemming from absorption of vibratory energy in oscillating systems can be conveniently viewed in an example of a system with one degree of freedom, consisting of the concentrated mass M, the element of elasticity C and the loss resistance R.

The free oscillations of the system arise with a sudden change in its condition. In this case the differential equation relative to diplacement of the mass $x=x(t)$ takes the form [44]

$$M\ddot{x}+R\dot{x}+Cx=0, \qquad (1.1)$$

where $\dot{x}=\frac{\partial x(t)}{\partial t}$; $\ddot{x}=\frac{\partial^2 x(t)}{\partial t^2}$; $R\dot{x}$ — frictional force

In the general case the loss resistance may have varying dependence on the frequency of oscillation of the mass. To begin with we will suppose that $R=R_1=\text{const}$. Then absorption of energy in the system will be proportional to the oscillating velocity $\dot{x}$, which corresponds to the so-called viscous, or liquid, friction. The solution to equation (1.1) $R=R_1$ is

$$\dot{x}=Ae^{j\dot{\omega}_{01}t}e^{-\delta_1 t}=Ae^{(j\dot{\omega}_{01}-\delta_1)t}, \qquad (1.2)$$

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where A is the amplitude of the oscillatory velocity of the mass, determined from the initial conditions; $\omega^{*}_{01}=\sqrt{\omega_0^2-\delta_1^2}$ is frequency of free damping of oscillations of the system when $R_1\neq 0$ и $\delta_1<\omega_0$; $\omega_0=\sqrt{CM^{-1}}$ is free damping of oscillations of the system in the absence in it of losses ($R_1=0$); $\delta_1=R_1/2M$ is constant of attenuation of the system.

From expression (1.2) it is evident that when $\omega_0>\delta_1$ (subcritical damping) with rise in loss resistance R_1 the free oscillations of the system are damped more quickly.

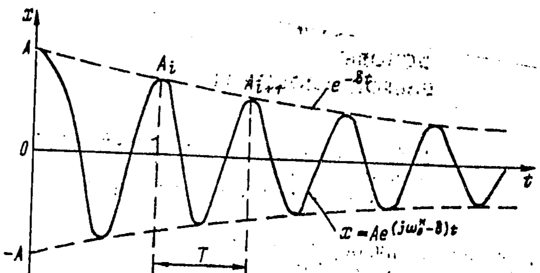

Fig. 1. Attenuating free oscillations in a system with one degree of freedom.

The frequency of oscillations therewith decreases insofar as R_1 impedes the movement of mass M.

The ratio of amplitudes of two adjacent maximums of attenuating oscillation (Fig. 1) is constant for a given R_1:

$$\frac{A_i}{A_{i+1}}=e^{\frac{2\pi\delta_1}{\overset{*}{\omega}_{01}}}=e^d, \qquad (1.3)$$

where d is the logarithmic decrement of oscillations;

$$d=\frac{2\pi\delta_1}{\overset{*}{\omega}_{01}}=\delta_1 T_1; \qquad (1.4)$$

T_1 is the period of oscillations.

The ratio of energy absorbed in the system during the period of oscillation T_1 to potential energy in the system is called the coefficient of absorption ψ. Since the potential energy in the system decreases somewhat during the period, it is advisable to relate the absorbed energy to the average value of the potential energy during the period. Taking the foregoing into account

$$\psi=\frac{2(w_i-w_{i+1})}{w_i+w_{i+1}}=e^d-e^{-d}, \qquad (1.5)$$

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where $w = A^2 e^{-2\delta_1 t} C/2$ is potential energy.

At low values of $d(d^2 \ll 6)$

$$\psi \approx 2d = \frac{4\pi\delta_1}{\omega_{01}^*}. \tag{1.6}$$

Quantity

$$\eta_0 = \frac{R_1}{\omega_0 M} = \frac{R_1\omega}{C} = \frac{2\delta_1}{\omega_0} \tag{1.7}$$

is called the loss factor of vibratory energy in an oscillating system. From comparison of (1.5) and (1.7) it follows that when $\omega^*_{01} \approx \omega_0 (d^2 \ll 6)$

$$\eta_0 \approx \frac{d}{\pi} \approx \frac{\psi}{2\pi}. \tag{1.8}$$

At low values of d [13]

$$\eta_0 = \frac{2d^2}{\sqrt{4\pi^2 + d^2}} \approx \frac{d}{\pi}(1 - 0{,}0127d^2). \tag{1.9}$$

The constants ω^*_{01} and δ_1, which characterize the attenuating oscillations of the system under study, in case R=const are expressed through the loss factor as follows:

$$\omega_{01}^* = \omega_0 \sqrt{1 - \frac{\eta_0^2}{4}}; \quad \delta_1 = \frac{\eta_0\omega_0}{2}. \tag{1.10}$$

It will be noted that $\omega^*_{01} \sim \omega_0 (1 - \eta_0^2/8)$ when $\eta_0^2 \ll 4$. From expressions (1.2) and (1.10) it is evident that when $\delta_1 > \omega_0$ ($\eta_0 > 2$ is supercritical damping) frequency ω^*_{01} takes on imaginary value, consequently movement of the system with such losses becomes aperiodic:

$$x = A\left[e^{-\left(\delta_1 - \sqrt{\delta_1^2 - \omega_0^2}\right)t} + \delta_1 t e^{-\left(\delta_1 + \sqrt{\delta_1^2 - \omega_0^2}\right)t}\right]. \tag{1.11}$$

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The loss factor value, which when exceeded the movement of the system become aperiodic, is called critical ($\eta_{кр}=2$).

Under internal friction, characteristic of deformable solids, R is inversely proportional to frequency

$$R=R_2=R_0\frac{\omega_0}{\omega^*_{02}}, \qquad (1.12)$$

where R_0 is value R_2 when $\omega^*_{02}=\omega_0$.

Solution of equation (1.1) when R=R leads to the result

$$\dot{x}=Ae^{\left(j\omega^*_{02}-\delta_2\right)t}, \qquad (1.13)$$

where ω^*_{02}, δ_2 are the frequency of free attenuating oscillations and constant of attenuation of the system, equal to:

$$\left.\begin{aligned}\omega^*_{02}&=\omega_0\sqrt{\frac{1+\sqrt{1-\eta_0^2}}{2}}\\ \delta_2&=\omega_0\sqrt{\frac{1-\sqrt{1-\eta_0^2}}{2}}\end{aligned}\right\}(\eta_0\ll 1) \qquad (1.14)$$

$$\omega^*_{02}=\delta_2=\omega_0\frac{\sqrt{\eta_0+1}-\sqrt{\eta_0-1}}{2} \quad (\eta_0\gg 1).$$

At low values of $\eta_0(\eta_0^2<<1)$

$$\omega^*_{02}\approx\omega_0\left(1-\frac{\eta_0^2}{8}\right);\qquad \delta_2=\frac{\omega_0\eta_0}{2}. \qquad (1.15)$$

In this case an increase of losses in the system also lowers the frequency of the free attenuating oscillations. With rise in loss factor ($\eta_0>>1$)

$$\omega^*_{02}=\delta_2\approx\frac{\omega_0}{2\sqrt{\eta_0}}. \qquad (1.16)$$

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Thus at R inversely proportional to frequency, there is no critical damping, insofar as ω_0 at any η_0 exceeds δ_2.

Substituting expression (1.13) into equation (1.1) shows that the term describing losses in the system becomes equal, consequently, the losses in this case are proportional to deformation of the element of elasticity x.

In some works, for example [13, 44], as losses in the system increase, where $R=1/\omega$, a rise in frequency of free attenuating oscillation is obtained. This result is derived in error, because $\dot{x}=j\omega^*_{02}x\sim j\omega_0 x$ is substituted into equation (1.1) instead of the obvious $\dot{x}=(j\overset{*}{\ } - \ )x$.

Herewith the solution to equation (1.1) when $R=R_3=R_0\frac{\omega^*_{03}}{\omega_0}$

$$\dot{x}=Ae^{\left(j\omega^*_{03}-\delta_3\right)}, \qquad (1.17)$$

where

$$\omega^*_{03}=\omega_0\sqrt{\frac{4}{4+\eta_0^2}};$$
$$\delta_3=\omega_0\sqrt{\frac{\eta_0^2}{4+\eta_0^2}}.$$

Specifically,

$$\omega^*_{03}\approx\omega_0\left(1-\frac{\eta_0^2}{8}\right),$$
$$\delta_3\approx\frac{\omega_0\eta_0}{2}$$

where $\eta_0^2\ll 4$, a $\omega^*_{03}\approx\omega_0\frac{2}{\eta_0}$, $\delta_3\approx\omega_0\left(1-\frac{2}{\eta_0^2}\right)$ where $\eta_0^2\gg 4$.

Thus at any η_0 the constant of attenuation δ_3 remains less than ω_0, therefore there will be no critical damping in this case either.

Fig. 2 depicts the dependence of the frequencies of attenuating oscillations and constants of attenuation on η_0 for the cases examined above.

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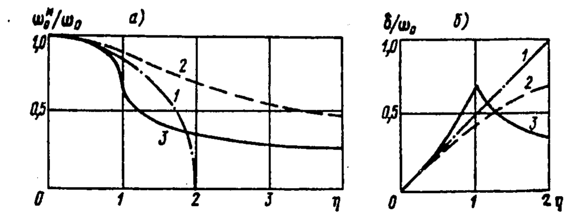

Fig. 2. Frequencies of free oscillations (a) and constants of attenuation (b) in a system with one degree of freedom depending on loss factor.

$1 - R=\text{const};\ 2 - R=R_0\frac{\omega}{\omega_0};\ 3 - R=R_0\frac{\omega_{03}}{\omega_0}.$

With increase in the loss factor the frequency of free attenuating oscillations in the system drops fastest of all where $R\equiv 1/\omega$, but where $R\equiv\omega$ the drop is slowed down. When $\eta_0<1$ decrease of this frequency and the constant of attenuation depend little on the character of frequency dependence of the loss resistance.

Induced oscillations of the system arise with exertion on mass M of force $F(t)=F \exp (j\omega t)$ (ω is the frequency of change in force, F_0 is its amplitude). The equation of system movement assumes the form [44]

$$M\ddot{x}+R\dot{x}+Cx=F_0e^{j\omega t}. \qquad (1.18)$$

Solution to this equation:

$$\dot{x}=\frac{F_0e^{j\omega t}}{|z_F|\,e^{j\varphi_z}}, \qquad (1.19)$$

where $z_F=R+j\,M+C/j$ is mechanical resistance of the system relative to force F; $\phi_z=\arccos\frac{R}{|z_F|}$ is the shift in phases between F and $\dot{x}$.

The frequency of the induced oscillations is equal to the frequency of the force exerted on the system. From expression (1.19) it follows that when $R=\text{const}$ $\dot{x}$ has the greatest value when $\omega=\omega_0$, the amplitude of which

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is equal to $\dot{x}_1=F_0/R=F_0/\omega_0\eta_0 M$. As frequency ω draws away from ω_0 the amplitude of the oscillatory velocity $\dot{x}$ drops. In connection with this the dependence of $\dot{x}$ on frequency is much like the frequency characteristic of an electrical selective filter (Fig.3).

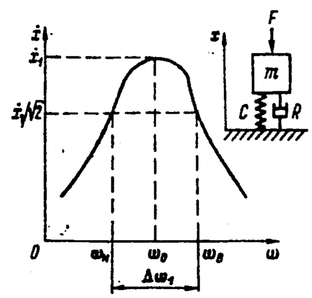

Fig. 3. Dependence of amplitude of oscillations of the system with one degree of freedom on frequency.

The dependence of $\ddot{x}$ and $\ddot{x}$ on frequency under induced oscillations of the system are much like the frequency characteristics of cut-off electrical filters which pass frequencies $\omega<\omega_0$ for x and $\omega>\omega_0$ for $\ddot{x}$. Therefore it is convenient to seek solution of the equation of movement of the oscillatory system relative to $\dot{x}$, since in this case the role of losses in the system will be shown more graphically.

By analogy of radio engineering we shall isolate the band of frequencies $\Delta\omega_1$, in which the amplitude of the oscillatory velocity of the system exceeds values $\dot{x}_1/\sqrt{2}$. The modulus of resistance of the system $Q\nu=\pm1$ may be written as [44]

$$|z_F|=R\sqrt{1+Q^2\nu}, \qquad (1.20)$$

where $Q=\frac{\omega_0 M}{R}=\frac{1}{\eta_0};\ \nu=\frac{\omega}{\omega_0}-\frac{\omega_0}{\omega}$ is the frequency difference coefficient. The values of frequencies ω_H and ω_B, which are bounds of $\Delta\omega_1$, are determined from condition $Q\nu=\pm1$. The difference in the values of the frequency difference coefficient which correspond to this condition is

$$\nu_B-\nu_H=\frac{2}{Q}=\frac{(\omega_B+\omega_H)(\omega_B-\omega_H)}{\omega_0^2}\approx\frac{2\Delta\omega_1}{\omega_0}. \qquad (1.21)$$

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Here $\Delta\omega_1=\omega_B-\omega_H$; it is assumed also that $\omega_B+\omega_H\sim 2\omega_0$. From expression (1.21) it follows that

$$Q\approx\frac{\omega_0}{\Delta\omega_1}\quad \text{или}\quad \eta_0=\frac{1}{Q}\approx\frac{\Delta\omega_1}{\omega_0}, \qquad (1.22)$$

therefore Q is the quality factor of the system while the loss factor at frequency ω_0 is the relative band of frequencies passed by the system with attenuation of no less that 0.707 of maximum.

Thus with increase in loss resistance of the system the amplitude of its oscillatory velocity decreases and the frequency band in which amplitude $\dot{x}$ exceeds value $\dot{x}_1/\sqrt{2}$ grows wider.

It was shown above that amplitude $\dot{x}$ where R=const assumes a greater value $\dot{x}$ when $\omega_1=\omega_0$. Not so with frequency-dependent R. If $R=R_0\omega_0/\omega$, then the greatest value of x will be at frequency

$$\omega_2=\omega_0\sqrt[4]{1+\eta_0^2}. \qquad (1.23)$$

When R=R / the greatest amplitude of x will be at frequency

$$\omega_3=\frac{\omega_0}{\sqrt[4]{1+\eta_0^2}}. \qquad (1.24)$$

It will be noted that values ω_2 and ω_3 are at variance with the corresponding values of frequencies of free attenuating oscillations ω^*_{02} and ω^*_{03}.

The frequency bands $\Delta\omega_2$ and $\Delta\omega_3$ when $R=R_0\omega_0/\omega$ and $R=R_0\omega/\omega_0$ are

$$\Delta\omega_2=\Delta\omega_3=\omega_0\sqrt{2\sqrt{1+\eta_0^2}-2} \qquad (1.25)$$

and, consequently, narrower in comparison with $\Delta\omega_1=\omega_0\eta_0$, which occurs at R=const. Amplitudes of oscillatory velocity of the system at frequencies ω_2(with $R=R_0\omega_0/\omega$) and ω_3 (with $R=R_0\omega/\omega_0$) will have values

$$\dot{x}_2=\dot{x}_3=\frac{F_0\eta_0}{R_0\sqrt{2\sqrt{1+\eta_0^2}-2}}. \qquad (1.26)$$

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These values are higher in comparison with $x_1=F_0/R_0$ when R=const.

Thus, the dependence of loss resistance of the system on frequency increases the maximum amplitude of oscillatory velocity of the system and compresses the passband of its frequency characteristics (given that at frequency ω_0 the loss resistance in all three cases is the same).

Energy $W_{\text{пог}}$, absorbed in the system with losses in one period of ascillation T, is equal to the work completed in this time. Assuming that

$$F=F_0\cos\omega t;\quad \dot{x}=\frac{F_0}{|z_F|}\cos(\omega t-\varphi_z), \tag{1.27}$$

after the obvious calculation we get

$$W_{\text{пог}}=\int\limits_{x(0)}^{x(T)}Fdx=\int\limits_0^T F\frac{dx}{dt}dt=\int\limits_0^T F\dot{x}dt=\frac{F_0^2}{2|z_F|}T\cos\varphi_z. \tag{1.28}$$

Since $T=2\pi/\omega$, but $\cos\phi_z=R/|z_F|$, then from expression (1.28) it follows that

$$W_{\text{пог}}=\pi A^2\omega R\equiv\eta, \tag{1.29}$$

where $A=F_0/\omega|z_F|$ is the amplitude of displacement of the mass of the system. From formula (1.29) it is evident that the energy absorbed in the system is directly proportional to the loss resistance or the loss factor.

From ratios of (1.27) it follows that

$$x=\frac{F_0}{\omega|z_F|}\cos\left(\omega t-\varphi_z-\frac{\pi}{2}\right)=\frac{F_0}{\omega|z_F|}\sin(\omega t-\varphi_z)=$$
$$=\frac{1}{\omega|z_F|}\left(\sqrt{F_0^2-F^2}\cos\varphi_z-F\sin\varphi_z\right). \tag{1.30}$$

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This equation links variables x and F. If we transfer to new quadrate coordinates x_1, F_1, turned in relation to x, F on an angle of $\Theta=45°$ (Fig. 4), then using the known ratios $x=x_1\cos\Theta+F_1\sin\Theta$ and $F=-x\sin+F\cos\Theta$ from equation (1.30) we get

$$x_1^2\frac{\omega^2|z_F|^2(1-\sin\varphi_z)}{F_0^2\cos^2\varphi_z}+F_1^2\frac{1+\sin\varphi_z}{F_0^2\cos^2\varphi_z}=1. \quad (1.31)$$

Equation (1.31) which links variables x and F is equation of the ellipse with semiaxes

$$a=\frac{F_0\cos\varphi_z}{\omega|z_F|\sqrt{1-\sin\varphi_z}}; \qquad b=\frac{F_0\cos\varphi_z}{\sqrt{1+\sin\varphi_z}}. \quad (1.32)$$

From (1.30) and (1.31) it follows that the trajectory of the coordinates in plane xF as force F is changed through one period of oscillation constitutes a closed loop in the form of an ellipse. The area of this ellipse is equal to

$$S_{\text{эл}}=\pi ab=\frac{\pi F_0^2\cos\varphi_z}{\omega|z_F|}=\pi A^2\omega R. \quad (1.33)$$

Comparison of (1.29) and (1.33) shows that the area of the ellipse described by equation (1.30) in coordinate plane xF, is equal to the energy absorbed in the system during the period of oscillation. The axes of the ellipse are turned in this plane at a 45° angle to coordinates x and F (see Fig. 4). In the absence of losses in the system the ellipse degenerates to a segment of a straight line.

Since the potential energy of the system is equal to $W_{\Pi OT}=CA^2/2$, then, taking (1.29) into account, the ratio of the energy absorbed during the period of oscillation in the system to $W_{\Pi OT}$ is

$$\frac{W_{\text{пог}}}{W_{\text{пот}}}=2\pi\eta, \quad (1.34)$$

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where η is the loss factor of the system at excitation frequency ω:

$$\eta=\frac{\omega R}{C}. \quad (1.35)$$

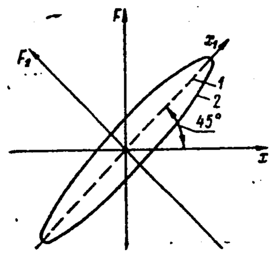

Fig. 4. Diagram of displacement of a system with one degree of freedom in the force - displacement plane.

$1-\eta=0;\ 2-\eta\neq0.$

It is not difficult to note the correlation between expression (1.35) and the analogous expression for free oscillations of the system (1.7).

In conclusion we note that at set induced oscillations of a system, excited by a force with a constant amplitude F_0 and frequency ω, the oscillatory velocity $\dot{x}$ and displacement x of the mass are proportional one to the other

$$\dot{x}=j\omega x. \quad (1.36)$$

Substituting parity (1.36) into equation (1.18) we get

$$M\ddot{x}+\overline{C}x=F=F_0e^{j\omega t}, \quad (1.37)$$

where $\overline{C}$ is the composite rigidity of the system

$$\overline{C}=C\left(1+j\frac{\omega R}{C}\right). \quad (1.38)$$

Taking (1.35) into account, we shall rewrite (1.38) as

$$\overline{C}=C(1+j\eta). \quad (1.39)$$

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Thus induced oscillations of a system with one degree of freedom is totally described by the composite rigidity in any character of dependence of loss resistance on frequency.

In systems where rigidity ($M\omega^2<C$, $\omega<\omega_\mu$), the ratio of force F to displacement x that it excites, as follows from equation (1.37) is

$$\frac{F}{x}=\overline{C}=C(1+j\eta). \tag{1.40}$$

If $\eta^2<1$, then $1+j\eta\sim e^{j\eta}$, consequently

$$\frac{F}{x}\approx e^{j\eta}. \tag{1.41}$$

From (1.41) it is evident that the loss factor η in systems, where rigidity predominates, constitutes a phase shift between the force and the displacement it excites. With a phase shift equal to zero ($\eta=0$), according to (1.34) no absorption of energy takes place in the system.

§ 2 Absorption of Vibratory Energy in Deformable Media

Absorption of vibratory energy in solid elastic media (rods, plates and other systems with distributed parameters) occurs with their deformation. The relationship between these deformations and the corresponding tensions is attributable to elastic constants. Therefore, such media constitute systems governed by elasticity (rigidity). Two types of deformation are inherenet in isotropic infinite media: expansion deformation and shear deformation, which are characterized by two elastic constants (Lamé's constants) λ and μ.

Complex deformation of a finite medium may be thought of as a combination of the two referenced simple deformations. Correspondingly the elastic constant which characterizes complex deformation may be expressed through Lamé's constants. For instance, Young's modulus for longitudinal deformations of a rod is equal to [44]:

$$E=\frac{\mu(3\lambda+2\mu)}{\lambda+\mu}. \tag{1.42}$$

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For dissipative media the Lamé's constants assume the complex form

$$\bar{\lambda}=\lambda_0(1+j\eta_\lambda);\quad \bar{\mu}=\mu_0(1+j\eta_\mu), \qquad (1.43)$$

where $\eta\lambda$ and $\eta\mu$ are the loss factors which define, in accordance with (1.41) the phase shift between tension and deformation. Usually $\eta\mu\sim\eta\lambda$ [65]. The elasticity constants for complex deformation of dissipative media also have a complex form. For example, with longitudinal oscillation of a rod made of isotropic material Young's modulus is expressed as

$$\bar{E}=E_0(1+j\eta_E), \qquad (1.44)$$

where $E_0=2\mu_0(1+\sigma)$,

$$\eta_E=\frac{(1+2\sigma^2)\,\eta_\mu+\sigma(1-2\sigma)\,\mu_\lambda}{1+\sigma}.$$

Where σ is Poisson's ratio. It will be recalled that it characterizes elastic properties of a medium. So for a liquid $\sigma=0.5$. The Poisson's ratios for rubber and rubber-like materials are almost the same, since these are much like a liquid in their elastic charateristics. For most metals used in shipbuilding $\sigma=0.29$

Table 1 shows experessions for loss factors in various types of complex deformations (longitudinal and flxural oscillations of rods and plates, torsional oscillations of rods) and the values of these factors at $\sigma=0.5$ and $\sigma=0.29$ [44]. As is evident the losses are governed mainly by shear deformation, and also that in all the listed deformations of plates and rods made from the same material the loss factors are practically equal.

The phase shift between tension and deformation in a dissipative medium and, consequently, the absorption of vibratory energy can depend on various physical phenomena. The basic ones are: viscous friction, mechanical hysteresis, elastic flow and relaxation of the material.

Viscous, or liquid, friction results from friction of particles of the substance on each other [44]. The greater the relative velocity of the particle flux the more substantial it is. Therefore, the loss factor under viscous friction is proportional to frequency, but loss resistance is not dependent upon frequency, as follows from expression (1.35).

Mechanical hysteresis is sometimes called solid friction [43] or internal losses [13]. When force is exerted on an elastic medium an irreversible

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Table 1

Loss Factors in Plates and Rods Under Various Deformations

Type of Deformation	Loss Factor	Loss Factor Values At $\sigma=0.5$	At $\sigma=0.29$
Torsional Oscillations of Rods	$\eta\mu$	$\eta\mu$	$\eta\mu$
Longitudinal and Flexural Oscillations of Rods	$\frac{(1+2\sigma^2)\eta\mu+\sigma(1-2\sigma)\eta\lambda}{1+\sigma}$	$\eta\mu$	$0.91\eta\mu+0.09\eta\lambda$
Longitudinal and Flexural Oscillations of Plates	$\frac{(1-2\sigma+2\sigma^2)\eta\mu+}{1-\sigma}$ $\frac{+\sigma(1-2\sigma)\eta\lambda}{1-\beta}$	$\eta\mu$	0.83 +0.17

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microchange of the structure (turning and disintegration of crystals in metal, breakdown of the molecular chain in plastics, etc) occurs in the medium. As a result, when tension is removed residual deformation forms in the medium, which in a periodic process causes deformation to lag in phase behind the corresponding tension. The relationship between them takes on a hysteretic character and the hysteresis constant is determined by the ratio of residual deformation to the maximum. Under small deformations the constant of hysteresis is equal to the loss factor, i.e. the angle between the vectors of tension and deformation.

At the ideal hysteresis its constant does not depend on time of the tension action and, therefore, the corresponding loss factor does not depend on frequency. Friction resistance is inversely proportional to frequency.

Elastic flow of the material is characterized in that residual deformation is proportional to time of exertion of the force. Time of the action of periodic force is proportional to the period, therefore, the value of residual deformation increases with decrease in frequency. The corresponding loss factr, equal to the ratio of residual deformation to the maximum, will be in this case inversely proportional to frequency.

Relaxation of material results from change in molecular structure, leveling of temperatures between sectors of the medium opposite in sign to deformation, etc. In a relaxing medium under constant deformation the resultant tension gradually subsides. As a result, a shift in phase occurs between tension and deformation and, consequently, absorbtion of vibratory energy also takes place.

The time necessary for establishment of tension in a relaxing medium is called relaxation time. Relaxation and the corresponding loss factor achieve greatest value at the frequency where oscillation period equals relaxation time. This frequency is called the relaxation frequency. In a deformable medium with relaxation of the material the modulus of its elasticity constant is dependent on frequency. With increase in frequency this dependence is characterized by a shift to higher values of the elasticity constant modulus near the relaxation frequency. Some materials, for instance rubber-like materials, can have a spectrum of relaxation frequencies.

Characteristic frequency relationships of elasticity constant moduli and loss factors corresponding to the subject moduli of vibratory energy absorption in deformable media are shown in Fig. 5.

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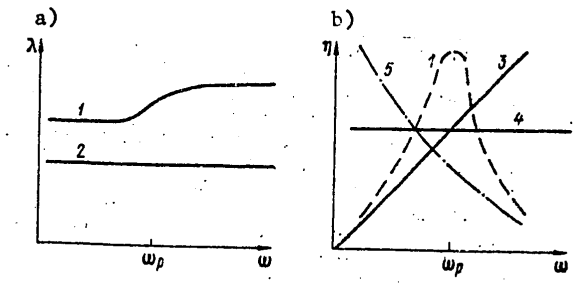

Fig. 5. Elasticity constant (a) and loss factor (b) of deformable media in dependence on frequency.

Key: 1. Relaxing medium; 2. Nonrelaxing medium; 3. Viscous friction; 4. Mechanical hysteresis; 5. Elasticity.

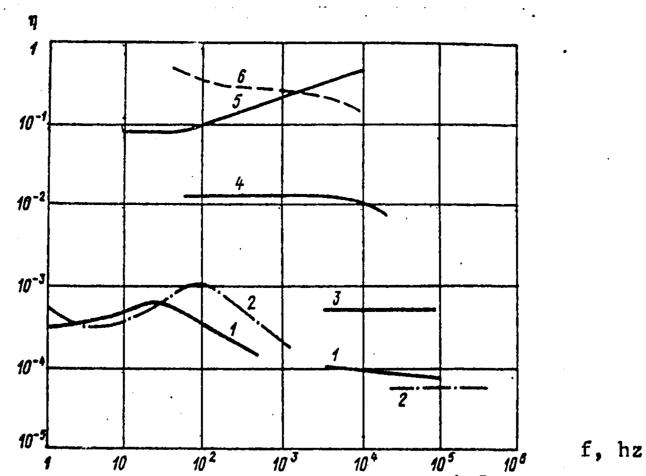

Fig. 6. Loss factors of various materials in dependence on frequency.

Key: 1. Steel; 2. Aluminum; 3. Copper; 4. Glass-plastic; 5. Rubber; 6. Plastic

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Worthy of attention are the values of loss factors of materials used in shipbuilding to make hull-frame structures and systems (steel, copper, aluminum, glass-plastic) as well as means of vibration absorption (Fig. 6). The values of loss factors of metals range approximately 10^{-4}-10^{-3}. From the form of the appropriate curves it may be concluded that at very low frequencies losses are attributable to elasticity of material (curve 2) and relaxation phenomena (curves 1 and 2). In this case relaxation occurs as a result of thrmal fluxes occurring along the thickness of the plates due to opposition in sign of deformations along both sides of the plane of the flexurally-oscillating plate. At high frequencies losses in the subject metals have a hysteretic origin. The greatest losses are in aluminum. Losses in glass-plastics are also the result of hysteresis (curve 4); they are an order of magnitude greater than losses in metals.

Rubbers and plastics have loss factors on the order of 0.1-1. The origin of these losses is of a relaxation character with given materials having an entire spectrum of relaxation frequencies. This explains the complex path of curves 5 and 6 (Fig. 6), which do not correspond to characteristic dependences of this type shown in Fig. 5.

The loss factor of metals and glass-plastic is practically independent of temperature. Plastics differ from metals in this respect. In its physio-mechanical properties plastic at low temperatures is much like glass. With an increase in temperature above a certain value, characteristic of the given plastic (the so-called vitrification temperature, dependent on frequency), plastic softens and changes to a rubbery substance, which with further increase in temperature can turn to liquid. This domain of change from glas-like to rubber-like condition is characterized by low losses.

§3. Dissipative Characteristics of Ship Machinery and Hull-Frame Structures.

Absorption of vibratory energy in ship structures, in the absence of means of vibration absorption, is primarily attributable to the following:
-- internal losses of energy in the construction material (see §2);
-- structural losses of energy due to the presence of welded seams, thermal insualtion, rivets, pipe and cable joints, etc;
-- losses of energy due to radiation of sound into the medium contiguous to plates of the hull-frame.

In practicable conditions it is difficult to establish a line of demarcation between these reasons. However, for evaluation of the dissipative properties of ship structures it is sufficient to know the total values of their loss factors.

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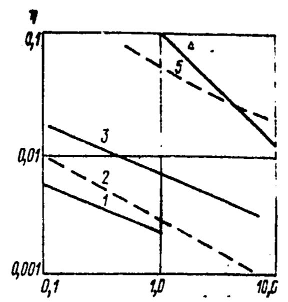

Fig. 7. Loss Factors in Ship Structures.

Key: 1. Steel vessel; 2. Steel vessel [62]; 3. Aluminum vessel; 4. Electric motor; 5. Electric motor (data of V.I. Popkov).

Figure 7 shows the frequency characteristics of energy loss factors in hull-frames of steel and aluminum ships [62, 34]. These factors have values on the order of 10^{-3}-10^{-2}. Losses in aluminum hull-frames have high values, probably due to the presence of riveted joints. The data shown are an order of magnitude higher than loss factors in ship hull-frame materials referenced in §2. It may be concluded in connection with this that losses in ship structures are primarily structural.

Loss factors measured in various ship structures (hull sheathing, bulkheads, decks) are practically the same. They depend little on displacement of the vessel [34]. However, as the hull is filled with equipment losses within it increase [63]. One can note the dependence of loss factors in hull-frame structures on their thickness. Thus for an aluminum thin-walled ship of riveted construction the loss factor at 1.0 khz is equal to 0.06. In the thick-walled welded hulls of steel ships the loss factor at this same frequency is lower by a factor of 2-3 (Fig. 7). It will be noted that higher loss factor values, on the order of 10^{-2}, are characteristic for the thin-walled bodies of automobiles, as pointed out in work [65].

Of practical interest are the losses in chassis of ship machinery. Figure 7 shows frequency characteristics of loss factors of two electric motors. The values of these factors are on the order of 0.01-0.1, i.e. an order of magnitude higher than loss factors of hull-frame structures. This is explained by the greater saturation of machinery with vibration absorbing elements (stator and rotor windings, insulation, etc.).

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The results presented show the possibility in principle of substantial increases in losses in ship hull-frame structures over the entire audio frequency spectrum and ship machinery at least at the higher frequencies.

Chapter 2. THE INFLUENCE OF VIBRATION ABSORPTION ON VIBROACOUSTICAL CHARACTERISTICS OF SHIP STRUCTURES

§4. The Energetics Method of Describing Vibroacoustical Characteristics of Ship Structures

From an acoustics point of view the hull-frame of a ship represents a combination of rigid plates reinforced by ribs and joined together in a certain way. The air spaces (compartments) enclosed between these plates (enclosures of the compartments) are acoustically linked with the latter and with each other. Therefore, ship structures, with the air spaces they enclose, may be viewed as a system of acoustical elements (plates and air spaces) acoustically joined together in accordance with the geometry of the ship's hull-frame.

An equation for the energy blance for each such element can be written by taking into account the arrival of energy from exterior sources, the exchange of energy between the connected elements and absorption of energy in the element. Solving a system of such equations, the number of which is equal to the number of elements making up the ship's hull-frame, allows one to evaluate the vibroacoustical characteristics of ship structures according to their dissipative properties. The energetics approach to solving similar problems is widely used in vibroacoustical engineering of structures such as buildings, aircraft, ships, etc. [34, 66, 118]. In spite of the number of approximations used in these methods, they allow one to derive comparatively simple analytical expressions which fit the experiment well.

It is most convenient to use the energetics method in conditions of diffuse fields, characterized by equal distribution of energy over the surface (volume) of the element. Then the energy of the element in a band of frequencies $\Delta\omega$ can be determined by the energy density w, the energy in the entire element being equal to wS in case of a plate with an area S and equal to wV in case of a compartment with volume V. It is assumed that there are several modes of oscillation of the element in the frequency band $\Delta\omega$. Simultaneous excitation of than no fewer than N modes of its oscillation is sufficient for existence of a diffuse field, where N=5 for a plate and N=10 for a volume. Taking the aforesaid into account, an equation for energy balance of element i of a ship's hull-frame consisting of n plates (compartment enclosures) and air spaces (compartments) can be written for a stationary process in the following form:

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$$W_i+\sum_{k=1}^{n}\alpha_{ki}c_k w_k-\sum_{k=1}^{n}\alpha_{ik}c_i w_i-\delta_i w_i=0, \qquad (2.1)$$

where W_i is energy arriving at element i from exterior sources; α_{ki} is the coefficient characterizing transmission of energy from element k to element i; c_k is the group velocity of waves in element k; σ_i is the ratio characterizing absorption of energy in element i.

The second term of the equation describes energy arriving a element i from elements joined to it. The third term describes the energy lost by element i due to its leakage into other elements and the fourth term describes the energy absorbed by element i.

Summation is performed in equation (2.1) for values k which correspond to elements directly adjoining element i. For the remaining values k $\alpha_{ki}=\alpha_{ik}=0$. It is obvious, in addition, that $\alpha_{ii}=0$.

The rational of equation (2.1) will be shown by two examples. As one of the examples let us look at a unidimensional structure in the shape of a strip divided into separate plates by transverse rigidity ribs (Fig. 8). We shall isolate from this structure a plate with index i-1 and a rib with index i. For these elements equation (2.1) may be written in the following form:

$$\begin{gathered}W_{i-1}+\alpha_{i-3,\,i-1}c_{i-3}w_{i-3}+\alpha_{i-2,\,i-1}c_{i-2}w_{i-2}+\alpha_{i+1,\,i-1}c_{i+1}+\\+w_{i+1}\alpha_{i,\,i-1}c_i w_i-(\alpha_{i-1,\,i-3}+\alpha_{i-1,\,i-2}+\alpha_{i-1,\,i+1}+\alpha_{i-1,\,i})\times\\\times c_{i-1}w_{i-1}-\delta_{i-1}w_{i-1}=0; \qquad (2.2)\\W_i+\alpha_{i-1,\,i}c_{i-1}w_{i-1}+\alpha_{i+1,\,i}c_{i+1}w_{i+1}-(\alpha_{i,\,i-1}+\alpha_{i,\,i+1})\times\\\times c_i w_i-\delta_i w_i=0.\end{gathered}$$

In accordance with [34], in these equations

$$\alpha_{jk}=\frac{<t_{jk}>_{\varphi}L_{jk}}{\pi};\quad \delta_k=\omega\eta_k S_k, \qquad (2.3)$$

where t_{jk} is the coefficient of the transfer of energy of the diffuse field of flexural waves from plate j to plate k; L_{jk} is length of the line joining plates j and k; $_k$ is the loss factor in plate k; S_k is the area of plate k. Coefficients $\alpha_{i-3,i-1}$, $\alpha_{i-1,i-3}$ $\alpha_{i-1,j+1}$, $\alpha_{i+1,i-1}$ define the transmission of energy from one plate to another through the rigidity rib, which is seen as an obstruction to the flexural waves.

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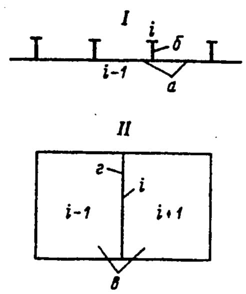

Fig. 8. Typical examples of ship structures.

Key: I. Plates (a), separated by rigidity ribs (б); II. Compartments (в), separated by a bulkhead (г).

In dependence on frequency the rigidity rib, according to [31], is viewed either as a vibration-inhibiting mass in which there are flexural waves with sectional transferance along the height of the rib (low frequencies), or as a plate with lateral (flexural) oscillations along its thickness (high frequencies). The cut-off frequency dividing the indicated regions of rib behavior is equal to [24]:

$$f_1 = \frac{0{,}08 c_{\text{пр}} h_{\text{р}}}{H_{\text{р}}^2},$$

where h_p is the thickness of the rigidity rib and H_p is its height. At frequencies higher than f_1 there is a possibility of an exchange of energy between the plates and the rigidity rib. At lower frequencies there is no such exchange, therefore, $\alpha_{i-1,\,i}=\alpha_{i+1,\,i}=\alpha_{i,\,i-1}=\alpha_{i,\,i+1}=\alpha_{i-2,\,i-1}=\alpha_{i-1,\,i-2}=0$ at frequencies $f<f_1$.

To simplifiy the problem of passage of energy of a diffuse field of flxural waves through a rigidity rib it is sometimes replaced by an articulated-supported line [31]. In this case the coefficients defining exchanges of energy between the rigidity rib and plates are equal to zero at all frequencies. Herein

$$\alpha_{i-1,\,i+1}=\alpha_{i+1,\,i-1}\approx 0{,}25\frac{L_{ik}}{\pi}\ [24].$$

It can be shown that with allowance for the exchange of energy between rigidity ribs and plates in the particular case the thickness of the plates and the rib, in the absence of losses in the latter, will be equal to

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$\alpha_{i-1,\,i+1}=\alpha_{i+1,\,i-1}=0{,}212\frac{L_{ik}}{\pi}$, which is near the approximate value indicated above.

Comparison of the coefficients with w in all terms of the equation (2.2) shows that

$$\eta_{jk}=\frac{\alpha_{jk}c_j}{\omega S_j} \tag{2.4}$$

has the character of the loss factor which defines the efflux of energy from plate j into plate k. Accordingly

$$\eta_{kj}=\frac{\alpha_{kj}c_k}{\omega S_k} \tag{2.5}$$

defines the influx of energy in $1/2\pi$ of the period into plate j from plate k. The coefficients η_{jk} and η_{kj} which define the echange of energy between the connected plates are in a fixed ratio. To determine this ratio let us examine the equation

$$\frac{\eta_{jk}}{\eta_{kj}}=\frac{\alpha_{jk}c_jS_k}{\alpha_{kj}c_kS_j}=\frac{\langle t_{jk}\rangle_\varphi c_jS_k}{\langle t_{kj}\rangle_\varphi c_kS_j}. \tag{2.6}$$

Value $\langle t_{jk}\rangle\phi$ for plates j and k which enter into compostion of the arbitrary number of plates p, according to [24] is equal to

$$\langle t_{jk}\rangle_\varphi=\frac{2}{\pi}\int\limits_0^{\varphi_{\kappa p\, j}} t_{jk}(\varphi)\,d\varphi, \tag{2.7}$$

where ϕ is the angle of incidence of flat flexural waves, which form a diffuse field, on the line of conjunction of the plates;

$$t_{jk}(\varphi)=t_{jk}\cos\varphi_j\cos\varphi_k, \tag{2.8}$$

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t_{jk} is the coefficient of the passage of energy the flexural waves with normal incidence on the line of conjunction of the plates, equal to

$$t_{jk}=\frac{2z_{\text{ом}\,j}\,z_{\text{ом}\,k}}{\left(\sum_{i=1}^{p} z_{\text{ом}\,i}\right)^2}, \qquad (2.9)$$

$z_{\text{ом}}=\frac{E_{\text{пл}} I_{\text{пл}} k_{\text{и пл}}}{\omega}$ — is characteristic resistance of a in relation to the bending moment; j and k are the angles of incidence and passage of the flat flexural wave through the line of conjunction of plates j and k; $\phi_{\text{кр}j}$ is the value of angle ϕj, which when exceeded angle ϕk becomes greater in comparison with $\pi/2 (\varphi_{\text{кр}j} \leqslant \pi/2)$.

A rough evaluation of the integral (2.7) can be made if change in cos ϕk is disregarded in expression (2.8). Taking such an allowance into account we get

$$\frac{\eta_{jk}}{\eta_{kj}} \approx \frac{c_j^2 S_k}{c_k^2 S_j}. \qquad (2.10)$$

This takes into account that in accordance with expression (2.9) $t_{jk}=t_{kj}$. It will be noted that the error in calculation (2.10) resulting from the noted assumption does not exceed 20% with any ratio of thicknesses of plates j and k.

The ratio (2.10) holds true also for two plates divided by a rigidity rib. In this $h_j=h_k$, consequently, $c_j=c_k$.

According to [34] the number of modes of flexural oscillations of a plate, whose basic frequencies fall in the frequency band $\Delta\omega$, is

$$N(\Delta\omega)=\frac{\omega\Delta\omega S}{\pi c^2}=n(\omega)\,\Delta\omega, \qquad (2.11)$$

where $n(\omega)$ is the density of the basic frequencies of the plate.

Taking expression (2.11) into account, ratio (2.10) may be rewritten as

$$\frac{\eta_{jk}}{\eta_{kj}} \approx \frac{n_k(\omega)}{n_j(\omega)}. \qquad (2.12)$$

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Thus the ratio of the coefficients η_{jk} and η_{kj} is inversely proportional to the ratio of the densities of the basic frequencies of the plates exchanging the energy. This can be explained in that each mode of one plate excites only the mode of the second plate which corresponds to it in frequency. Therefore, the energy emitted from the plate with the lowest number of modes exceeds the energy emitted in the opposite direction.

Another example, characteristic of ship conditions, which will clarify the rationale of equation (2.1) can be shown by two adjoining compartments separated by a bulkhead. If these compartments are assigned indices i- and i+ and the bulkhead is assigned i, then the equations describing the values of energy in these two elements will appear as:

$$\begin{gathered}
W_{i-1}+\alpha_{i,\,i-1}c_i w_i+\alpha_{i+1,\,i-1}c_{i+1}w_{i+1}-\alpha_{i-1,\,i}c_{i-1}w_{i-1}-\\
-\alpha_{i-1,\,i+1}c_{i-1}w_{i-1}-\delta_{i-1}w_{i-1}=0;\\
W_i+\alpha_{i-1,\,i}c_{i-1}w_{i-1}+\alpha_{i+1,\,i}c_{i+1}w_{i+1}-\alpha_{i,\,i-1}c_i w_i-\\
-\alpha_{i,\,i+1}c_i w_i-\delta_i w_i=0;\\
W_{i+1}+\alpha_{i-1,\,i+1}c_{i-1}w_{i-1}+\alpha_{i,\,i+1}c_i w_i-\alpha_{i+1,\,i-1}c_{i+1}w_{i+1}-\\
-\alpha_{i+1,\,i}c_{i+1}w_{i+1}-\delta_{i+1}w_{i+1}=0.
\end{gathered} \tag{2.13}$$

Here $c_{i-1}=c_{i+1}=c_0$ is the velocity of sound in air; $c_i=2c_{\text{И пл}}$ is the group velocity of flexural waves in the plate from which the bulkhead is made. The coefficients

$$\begin{gathered}
\alpha_{i-1,\,i+1}=\frac{\omega V_{i-1}}{c_0}\eta_{i-1,\,i+1};\\
\alpha_{i+1,\,i-1}=\frac{\omega V_{i+1}}{c_0}\eta_{i+1,\,i-1},
\end{gathered} \tag{2.14}$$

define the transmission of energy through the bulkhead owing to nonresonant oscillations of its modes. Here V is the volume of the compartment. The values of coefficients can be expressed through sound insulation of the bulkhead ЗИ₀, determined by the law of mass [7]:

$$\alpha_{i+1,\,i-1}=\alpha_{i-1,\,i+1}=\frac{S}{4ЗИ_0}, \tag{2.15}$$

where S is the area of the bulkhead;

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$$3И_0 = \frac{\pi m_{пл}^2 f^2}{\rho_0^2 c_0^2};$$

$m_{ПЛ}$ is the mass of the bulkhead per unit of area.

Coefficients $\alpha_{i,i+1}, \alpha_{i+1,i}, \alpha_{i-1,i}$ и $\alpha_{i,i-1}$ define the exchange of energy between the air spaces and the bulkhead, if resonant oscillations occur within it. In the absence of these ($f_{И1}$ $f_В$; $f_{И1}$ is the first resonant frequency of flexural oscillations of the bulkhead, $f_В$ is the upper limit of the subject frequency spectrum) these coefficients equal zero. At $f_И$ $f_В$ in the frequency spectrum $f_{И1} \div f_В$ transmission of energy from compartment to the other is possible due to excitation of resonant oscillations of the bulkhead by the incident sound and reradiation of this energy by the bulkhead. It is obvious that

$$\eta_{i,i-1} = \eta_{i,i+1} = \eta_{изл} \tag{2.16}$$

are loss factors in the bulkhead due to sonic radiation by one of its sides. According to the determination in [87]

$$\eta_{изл} = \frac{R_{изл}}{\omega m_{пл} S}, \tag{2.17}$$

where $R_{ИЗЛ}$ is the radiation resistance of the bulkhead, which can be calculated from data in works [8, 87].

The values of coefficients $\alpha_{i,i-1}$ и $\alpha_{i,i+1}$ are determined as

$$\alpha_{i,i-1} = \alpha_{i,i+1} = \frac{\omega S}{2c_{и\,пл}} \eta_{изл} = \frac{R_{изл}}{2c_{и\,пл} m_{пл}}. \tag{2.18}$$

Ratio (2.12) also holds true for the subject system, as is shown in work [86]. Taking this into account, the expression can be written as:

$$\eta_{i-1,i} = \eta_{i,i-1} \frac{n_i(\omega)}{n_{i-1}(\omega)}; \quad \eta_{i+1,i} = \eta_{i,i+1} \frac{n_i(\omega)}{n_{i+1}(\omega)}. \tag{2.19}$$

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Here $n_i(\ )$ can be determined from formula (2.11). According to [43]

$$n_{i-1}(\omega)=\frac{\omega^2 V_{i-1}}{2\pi^2 c_0^3};\qquad n_{i+1}(\omega)=\frac{\omega^2 V_{i+1}}{2\pi^2 c_0^3}. \quad (2.20)$$

Corresponding expression for coefficients $\alpha_{i-1,i}$, $\alpha_{i+1,i}$ are as follows:

$$\alpha_{i-1,\,i}=\alpha_{i+1,\,i}=\frac{\pi c_0^2 S}{2c_{\text{в пл}}^2}\eta_{\text{изл}}. \quad (2.21)$$

Coefficients δ in this case are equal to

$$\delta_{i-1}=\omega V_{i-1}\eta_{i-1};\quad \delta_i=\omega S\eta_i;\quad \delta_{i+1}=\omega V_{i+1}\eta_{i+1}. \quad (2.22)$$

Here $\eta_i=\eta_{\text{BH}}$ is the internal loss factor in the bulkhead with allowance made for the possible presence on it of a vibroabsorptive coating. Loss factors i-1 and i+1, which define absorption of energy in the corresponding compartments, can be expressed through reverberation time T

$$\eta_{i-1}=\frac{2{,}2}{fT_{i-1}};\qquad \eta_{i+1}=\frac{2{,}2}{fT_{i+1}}. \quad (2.23)$$

By solving the system of equations (2.1), compiled for all elements of a ship's hull-frame, an expression can be derived from the density of energy in any compartment to calculate the levels of air noise on a ship with a given distribution of vibratory energy sources. Such a method is more general in comparison to methods of determining levels of vibrations of compartment enclosures with subsequent calculation of air noise levels within the compartment, as is done in works [8, 34]. These works do not take into account the interaction of ship structures with air spaces along the sonic vibration propagation path. Therefore, the suggested method should be even more precise.

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§5. Vibroexcitability of Structures

Operating machinery and sonic pressure of air noise can be sources of vibration in ship structures.

For examination of the first case let us turn to equations (2.13). We will suppose that a vibrating machine is situated on bulkhead i and, consequently, $W_{i-1} = W_{i+1} = 0$, a $W_i = W_0$ (W_0 is the vibrational power of the source installed on the bulkhead). For simplicity we shall assume that the compartments, separated by the bulkhead, are identical. Inserting i=2 from equation (2.13) we get

$$W_0 + 2\alpha_{32} c_0 w_3 - 4\alpha_{23} c_{\text{н пл}} w_2 - \omega_2 S_2 \eta_{\text{вн}} w_2 = 0;$$
$$2\alpha_{23} c_{\text{н пл}} w_2 - \alpha_{32} c_0 w_3 - \delta_3 w_3 = 0. \qquad (2.24)$$

Having solved this system relative to w_2, we get

$$w_2 = \frac{W_0}{\omega S_2 \left(\eta_{\text{вн}} + 2\eta_{\text{нзл}} \frac{\mu}{1+\mu}\right)}, \qquad (2.25)$$

where $\mu = \frac{\eta_3 n_3(\omega)}{\eta_{\text{нзл}} n_2(\omega)}$; the remaining notations are same as in the preceeding paragraph. Value $n_3(\omega)/n_2(\omega)$ is proportional to frquency. Therefore, at sufficiently low frequencies ($\mu \to 0$)

$$w_2 \to \frac{W_0}{\omega S_2 \eta_{\text{вн}}}, \qquad (2.26)$$

but a high frequencies ($\mu \to \infty$)

$$w_2 \to \frac{W_0}{\omega S_2 (\eta_{\text{вн}} + 2\eta_{\text{нзл}})}. \qquad (2.27)$$

The difference in behavior of the system at low and high frequencies is explained by an increase in vibroexcitability of the bulkhead by air noise in the compartment with decrease in frequency. Therefore, almost all the energy radiated by the bulkhead is returned to it and absorption of energy in the system is governed only by internal losses in the bulkhead itself

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The coefficient $\eta_{изл}$ is usually on the order of 10^{-3}. In a hip bulkhead $\eta_{вн}$ has a value on the order of 10^{-2} and more; therefore, expression (2.26) holds true for all audio frequencies. Thus the density of vibratory energy in the bulkhead, which is excited by the source installed on it, is inversely proportional to its loss factor.

When the bulkhead is excited by sonic pressure of air noise we shall consider that its source is situated in the right compartment (see Fig. 8). Then $W_{i-1}=W_i=0$, a $W_{i+1}=W_0$. From equation (2.13) for this case, disregarding the energy transmitted into the adjacent compartment through the bulkhead, we have (i-1, i=2, i+3=3):

$$\begin{aligned}&\alpha_{31}c_0w_3-2\alpha_{23}c_{н\,пл}w_2-\delta_2w_2=0;\\&W_0+2\alpha_{23}c_{н\,пл}w_2-\alpha_{32}c_0w_3-\delta_3w_3=0.\end{aligned} \tag{2.28}$$

Having solved this system relative to w_2, we get

$$w_2=\frac{W_0}{\omega S_2\left(\eta_{вн}+\frac{\eta_3}{\mu_n}\cdot\frac{\eta_{изл}+\eta_{вн}}{\eta_{изл}}\right)}, \tag{2.29}$$

where $\mu_n=\frac{\bar{n}_2(\omega)}{n_3(\omega)}$.

At low frequencies the acoustical link between the bulkhead and the air space of the compartment is strong, $\mu_n\to\infty$ and

$$w_2\to\frac{W_0}{\omega S_2\eta_{вн}}. \tag{2.30}$$

At high frequencies, where the acoustical link weakens, $\mu_n\to 0$ and

$$w_2\to\frac{W_0\mu_n}{\omega S_2\eta_3}\,\frac{\eta_{изл}}{\eta_{изл}+\eta_{вн}}\to 0. \tag{2.31}$$

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The amplitude of vibrations excited in the bulkhead by sonic pressure of air noise is less as the internal loss factor within is greater. This holds true for all frequencies, including high frequencies, since for ship structures usually $\eta_{ВН} > \eta_{ИЗЛ}$.

From the first equation (2.28) one can derive the relative vibratory velocity of the plate ξ and the sonic pressure in compartment p which excites its. Taking into account that

$$w_2 = m_{пл} \langle \dot{\xi}^2 \rangle; \qquad w_3 = \frac{\langle p^2 \rangle}{\rho_0 c_0^2}, \qquad (2.32)$$

we get a known expression [87]

$$\frac{\langle \dot{\xi}^2 \rangle}{\langle p^2 \rangle} = \frac{\mu_n V}{S m_{пл} \rho_0 c_0^2} \frac{\eta_{изл}}{\eta_{изл} + \eta_{вн}}, \qquad (2.33)$$

where V is the volume of the compartment; S is the area of the bulkhead.

§6. Propagation of Vibrations Through Structures

Most characteristic for ship structures is a plate reinforced by periodically installed rigidity ribs. To examine the influence of damping on propagation of vibrations in such a structure let us turn to equations (2.2). We shall seek the ratio between densities of vibratory energy in two adjacent cells of a unidimensional ribbed plate, assuming that there are no sources of vibration within them. For simplicity let us assume that resonant oscillations of the rigidity ribs is also absent, consequently,

$$\begin{gathered} W_i = W_{i+1} = 0; \quad \alpha_{i-2,\, i-1} = \alpha_{i,\, i-1} = \alpha_{i-1,\, i-2} = \\ = \alpha_{i-1,\, i} = \ldots = 0; \qquad (2.34) \\ \alpha_{i-3,\, i-1} = \alpha_{i+1,\, i-1} = \alpha_{i-1,\, i-3} = \alpha_{i-1,\, i+1} = \ldots = \alpha_0; \\ c_{i-1} = c_{i-3} = \ldots = 2c_{к\,пл}. \end{gathered}$$

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Taking (2.34) into account, from the first equation (2.2) we get

$$2\alpha_0 c_{\text{н пл}} w_{i-3} + 2\alpha_0 c_{\text{н пл}} w_{i+1} - 4\alpha_0 c_{\text{н пл}} w_{i-1} - \\ - \dot{\omega}\eta_{\text{пл}} l_0 w_{i-1} = 0, \tag{2.35}$$

where l_0 is the distance between rigidity ribs and the coefficient α_0 is assigned to a unit of length of the rigidity rib.

Let us denote

$$\frac{w_{i-3}}{w_{i-1}} = \frac{w_{i-1}}{w_{i+1}} = \gamma. \tag{2.36}$$

Then expression (2.35) can be rewritten as

$$\gamma^2 + 1 + (2 + \chi)\gamma = 0, \tag{2.37}$$

where $\chi = \frac{\omega l_0 \eta_{\text{пл}}}{2c_{\text{н пл}}\alpha_0}$.

Solution to equation (2.37) is

$$\gamma = 1 + \frac{\chi}{2} \pm \frac{\chi}{2}\sqrt{1 + \frac{4}{\chi}}. \tag{2.38}$$

The plus sign applies to the case when the source of vibration is situated to the right of the subject cells and the minus sign when it is to the left.

If the square root in formula (2.38) is presented as a series then we get

$$\gamma = 1 \pm \chi^{\frac{1}{2}} + \frac{\chi}{2} \pm \frac{\chi^{\frac{3}{2}}}{8} + \ldots \tag{2.39}$$

The first three terms of this series conincide with expansion of the exponential function

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$$e^{\pm\sqrt{\chi}} = 1 \pm \chi^{\frac{1}{2}} + \frac{\chi}{2} \pm \frac{\chi^{\frac{3}{2}}}{6} + \dots \qquad (2.40)$$

Thus, with accuracy up to the difference in sums of the fourth and final terms of the series (2.39) and (2.40), one can write

$$\gamma = e^{\pm\sqrt{\chi}}. \qquad (2.41)$$

It is shown in work [24] that with accuracy up to 0.5 expression (2.41) holds true for conditions practically encountered in ship structures.

From expression (2.41) it follows that the density of vibratory energy in the adjacent cell can be determined as

$$w_{i-3} = w_{i-1} e^{\pm\sqrt{\chi}}. \qquad (2.42)$$

For evaluation of density of energy in further removed cells formula (2.42) can be generalized as

$$w(x) = w_0 e^{\pm\sqrt{\chi}\frac{x}{l_0}} = w_0 e^{\pm\sqrt{\chi_0}\,x}, \qquad (2.43)$$

where $\chi_0 = \dfrac{\omega\eta_{пл}}{2c_{н\,пл} l_0 \alpha_0}$.

From expression (2.43) it can be seen that an increase in the loss factor in a ribbed plate leads to a decrease in the density of energy and amplitude of vibrations propagating through the plate Expression (2.43) also constitutes resolution of the equation derived for density of vibratory energy in ribbed plates by methods much like those used in the theory of heat conductivity [24]. Herein lies the community of these methods with Westphal's methods used above.

§7. Sonic Radiation of Structures

Let us assume that a bulkhead between two identical compartments, excited by a source of vibration with power W_0, radiates sonic energy into both compartments (see Fig. 8). In accordance with equations (2.13) we have for this case

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$$\begin{aligned} &2\alpha_{23}c_{\text{н пл}}w_2 - \alpha_{32}c_0w_3 - \delta_3 w_3 = 0; \\ &W_0 + 2\alpha_{32}c_0w_3 - 4\alpha_{23}c_{\text{н пл}}w_2 - \delta_2 w_2 = 0. \end{aligned} \tag{2.44}$$

Having solved this system of equations relative to the sought value $w_1 = w_3$, we get

$$w_1 = w_3 = \frac{W_0}{\omega V_3 \left(\mu_n \eta_{\text{вн}} + \eta_3 \frac{2\eta_{\text{изл}} + \eta_{\text{вн}}}{\eta_{\text{изл}}}\right)}. \tag{2.45}$$

Notations here are the same as in §5.

At low frequencies the acoustical link between bulkhead and compartment is strong, inasmuch as the number of modes in these elements due to varying dependence on frequency $n_2(\omega)$ and $n_3(\omega)$ becomes comparable. Therefore, at the indicated frequencies ($\mu_n \to \infty$)

$$w_3 \to \frac{W_0}{\omega V_3 \mu_n \eta_{\text{вн}}}. \tag{2.46}$$

At high frequencies the number of modes in the compartment increase sharply in comparison to the number of modes in the bulkhead and the acoustical link between them weakens ($\mu_n \to 0$) and

$$w_3 \to \frac{W_0}{\omega V_3 \eta_3} \frac{\eta_{\text{изл}}}{2\eta_{\text{изл}} + \eta_{\text{вн}}}. \tag{2.47}$$

An increase in losses in the bulkhead decreases its sonic radiation into the compartment. Calling attention to itself is the identical dependence on $\eta_{\text{ВН}}$ of energy radiated by the bulkhead and the energy excited in the bulkhead by an external source [compare expressions (2.46) and (2.27)]. This points to the fact that the decrease in sonic radiation of the bulkhead when it is damped is explained by a decrease in its vibrations. In practical use of expression (2.45) one should keep in mind the change in $\eta_{\text{ИЗЛ}}$, which takes place with damping of a sound insulating plate, due to the change in ratio of contribution to this radiation of low frequency and resonant modes of oscillation, dependent to varying degrees on loss factors of the plate. This is addressed in more detail in §28.

From the first equation (2.44) one can determine the relationship of oscillatory velocity of the bulkhead ξ and sonic pressure p in the compartment, governed by radiation of the bulkhead. Taking into account the dependences (2.32) we have

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$$\frac{\langle \xi^2 \rangle}{\langle p^2 \rangle} = \frac{V_3}{S_2 m_{\text{пл}} \rho_0 c_0^2} \frac{\mu_n \eta_{\text{изл}} + \eta_3}{\eta_{\text{изл}}}. \qquad (2.48)$$

This formula conincides with the analogous, but less precise, ratio derived in work [66], at high frequencies ($\mu_n \to 0$). The ratio of ξ and p does not depend on losses in the bulkhead, which once again confirms the foregoing explanation of the reason for a decrease in sonic radiation of a flexurally-oscillating bulkhead with an increase of its internal losses.

§8. Sound Insulation of Structures

Let us place a noise source with an audio power of W_0 in compartment i-1 and determine the sound insulation ЗИ of bulkhead i, which separates compartments i-1 and i+1 (see Fig. 8). We shall seek the value of ЗИ in the form (i-1=1, i=2, i+1=3):

$$ЗИ = \frac{w_1}{w_3} = \frac{p_1^2}{p_3^2}. \qquad (2.49)$$

For this case two equations are sufficient, since we are interested in the relationship of energy densities. Therefore, equations (2.13) for plate i=2 and compartment i+1=3 will be written as

$$\begin{aligned} &\alpha_{13} c_0 w_1 + \alpha_{32} c_0 w_3 - 2\alpha_{21} c_{\text{и пл}} w_2 - 2\alpha_{23} c_{\text{и пл}} w_2 - \delta_2 w_2 = 0; \\ &\alpha_{13} c_0 w_1 + 2\alpha_{23} c_{\text{и пл}} w_2 - \alpha_{31} c_0 w_3 - \alpha_{32} c_0 w_3 - \delta_3 w_3 = 0. \end{aligned} \qquad (2.50)$$

Solution of the system (2.50) of equations relative to ЗИ is

$$ЗИ = \frac{V_3}{V_1} \frac{\eta_{13} \frac{n_1}{n_3} + \eta_3 + \eta_{\text{изл}} \frac{n_2}{n_3} \frac{\eta_{\text{изл}} + \eta_{\text{вн}}}{2\eta_{\text{изл}} + \eta_{\text{вн}}}}{\eta_{13} + \frac{n_2}{n_1} \frac{\eta_{\text{изл}}^2}{2\eta_{\text{изл}} + \eta_{\text{вн}}}}. \qquad (2.51)$$

This expression coincides with the analogous formula derived in work [66] if it is assumed that $\alpha_{23}=0$.

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Sound insulation of the bulkhead will be appreciable if the energy which has passed through it is entirely or almost entirely absorbed in the insulated compartment. To achieve this it is necessary that sound absorption in the given compartment be sufficiently great. To ensure such a situation these inequalities must be satisfied

$$\eta_3 \gg \eta_{13}\frac{n_1}{n_3}; \quad \eta_3 \gg \eta_{изл}\frac{n_2}{n_3}\frac{\eta_{изл}+\eta_{вн}}{2\eta_{изл}+\eta_{вн}}. \qquad (2.52)$$

From analysis of expression (2.51) it is not difficult to conclude that when the inequalities (2.52) are not met the value of sound insulation approaches 1.

Taking the inequalities (2.52) into account, and also keeping in mind that in ship conditions usually $\eta_{ИЗЛ} \ll \eta_{ВН}$, the formula (2.51) may be rewritten

$$ЗИ \approx \frac{V_3}{V_1}\frac{\eta_3}{\eta_{13}+\frac{n_2}{n_1}\frac{\eta^2_{изл}}{\eta_{вн}}}. \qquad (2.53)$$

From formula (2.53) it follows that with a sufficiently high loss factor in the bulkhead $\eta_{ВН}$

$$ЗИ \approx \frac{V_3\eta_3}{V_1\eta_{13}} = \frac{4\delta_3}{c_0 S}ЗИ_0, \qquad (2.54)$$

consequently, the sound insulation of the bulkhead is defined by value $ЗИ_0$, i.e. according to the law of mass (see §4.).

For anlysis of the influence of $\eta_{ВН}$ on sound insulation of the bulkhead let us examine the ratio of terms in the denominator of expression (2.53). The frequency dependences of these terms are graphically depicted in Fig. 9. Coefficient η_{13} takes into account formula (2.23) on the supposition that reverberation time T_3 in compartment i+1=3 is independent of frequency. The second term is defined by using formulas (2.17), (2.11) and (2.20). and also typical frequency dependence of plate radiation resistance $R_{ИЗЛ}$ [87]. From Fig. 9 it is evident that the second term can exceed the first only near the critical frequency of the bulkhead $f_{КР}$ (at this frequency $c_0=c_{И\ ПЛ}$), with the indicated excess being more substantial as the $\eta_{ВН}$ is less. The value of sound

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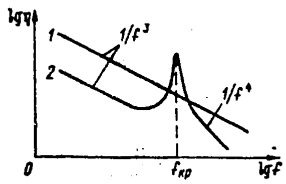

Fig. 9. Dependences of terms of expression (2.53) on frequency.

$1 - \eta_{н}$; $2 - \frac{n_2}{n_1}\eta^2_{изл}/\eta_{вн}$.

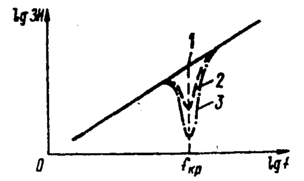

Fig. 10. Dependence of sound insulation of bulkhead ЗИ on frequency.

$1 - \eta_{вн2} \to \infty$; $2 - \eta_{вн2}$; $3 - \eta_{вн3} \to 0$ $(\eta_{вн3} < \eta_{вн2})$.

insulation of the bulkhead is less that the value determined by the law of masses, only near $f_{КР}$. We note that with a sufficiently high $\eta_{ВН}$ the indicated decrease may not even exist.

Chapter 3. VIBROABSORPTIVE COATINGS FOR SHIP STRUCTURES

§9. Methods of Determining Losses of Vibrational Energy in Oscillating Laminated Media

Vibroabsorptive coatings are applied to a ship structure plate which is to be damped and consists, as a rule, of several layers of various materials. Some vibroabsorptive construction materials also consist of several layers. In both cases we are concerned with a system of n layers (including the damped plate), the loss factor of which must be determined. The exact solution of this problem can be obtained on the basis of examination of energy of elastic deformations which take place in the layers. There are also approximate methods known for calculating loss factors in a laminated medium, some of which will be examined below.

The Deformation Energy Method. The loss factor of a laminated medium, as any other oscillatory system, can by analogy to (1.34) be written as

$$\eta_{\Sigma} = \frac{W_{пог\,\Sigma}}{\pi W_{пот\,\Sigma}}, \qquad (3.1)$$

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where $W_{\text{пог}\,\Sigma}$ is the total vibratory energy, absorbed in all layers of the system during half the period; $W_{\text{пот}\,\Sigma}$ is the total ptential energy of the system.

With deformation of an elastic dissipative medium having elastic constant λ the frictional resistance in it, by analogy to (1.35) is

$$R=\frac{\eta\lambda}{\omega}. \qquad (3.2)$$

The frictional force in a unit of volume of such a medium with relative deformation ε is equal to Rε, while energy absorbed in a unit of time is $R\varepsilon\varepsilon/2=R\varepsilon^2/2$. Energy absorbed in half the period, taking (3.2) into account, is equal to

$$W_{\text{пог}}=\frac{R\dot{\varepsilon}^2}{4f}=\frac{\pi\eta\lambda\varepsilon^2}{2}. \qquad (3.3)$$

Since the potential energy in a unit of volume of a medium is equal to $W_{\text{пот}}=\lambda\varepsilon^2/2$, equation (3.3) can be presented as

$$W_{\text{пог}}=\pi\eta W_{\text{пот}}. \qquad (3.4)$$

Taking formula (3.4) into account, expression (3.1) can be rewritten as

$$\eta_\Sigma=\frac{\sum\limits_{i=1}^{n}\sum\limits_{k=1}^{m}\eta_{ik}W_{\text{пот}\,ik}}{\sum\limits_{i=1}^{n}\sum\limits_{k=1}^{m}W_{\text{пот}\,ik}}. \qquad (3.5)$$

In this expression summation is performed for all m types of deformation which takes place in all n layers.

Both flexural and longitudinal oscillations can take place in ship structures [34]. Over and above flexural and longitudinal waves, these oscillations, related to longitudinal and lateral displacement of the plate surface, also excite transverse waves and compression waves in the joined layers, which propagate perpendicular to the plane of the layers. Thus there may be several types of deformation in the laminated medium in question.

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With flexural oscillations of a layer with flexural rigidity B its potential energy, falling with a unit of its surface, is equal to [24]:

$$W_{\text{пот и}}=\frac{1}{2}B|\theta'|^2, \qquad (3.6)$$

where θ is amplitude of the deflection angle of a layer section; the prime ´ denotes a derivative of the coordinate along which the flexural wave propagates. For longitudinal oscillations of a layer with tensile rigidity D [24]

$$W_{\text{пот п}}=\frac{1}{2}D|\zeta'|^2, \qquad (3.7)$$

where ζ is amplitude of the displacement of a cross section of a layer along the direction of propagation of the longitudinal wave.

The potential energy in a layer, in which a transverse wave propagates along the thickness h, is equal (for a unit of surface of the layer) to

$$W_{\text{пот с}}=\frac{1}{2}G\int_0^h|v'(z)|^2dz, \qquad (3.8)$$

where ν´(z) is distribution of shear deformation along coordinate z, directed along the thickness of the layer. For the case of propagation of a compression wave along the thickness of the layer

$$W_{\text{пот сж}}=\frac{1}{2}K\int_0^h|\zeta'(z)|^2dz, \qquad (3.9)$$

where K is the modulus of compressibility of the layer's material.

If the thickness of the layer is small in comparision to the length of the transverse and compression wave then expressions (3.8) and (3.9) assume the form

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$$W_{\text{пот с}} = \frac{1}{2} Gh \, |\nu'|^2;$$

$$W_{\text{пот сж}} = \frac{1}{2} Kh \, |\zeta'|^2. \qquad (3.10)$$

Here ν and ζ are amplitude values for the corresponding deformations.

Equation (3.5) holds true also for rod systems, consisting of n elements with identical cross section along their length. In this case in formulas (3.6) and (3.7) B and D are respectively flexural and tensile rigidity of an element in the rod system. In formulas (3.8) and (3.9) integration is performed along a cross section of the element, and in formula (3.10) the area of this section must be inserted instead of h.

The Complex Rigidity Method. With a thickness of layers much smaller than the wavelength of possible deformations, oscillation of the laminated medium are described by one wave number common to all layers. If one or several of the layers is made of dissipative material, the rigidity of the laminated medium, corresponding to the type of oscillation taking place within it, assumes a complex character. With flexural oscillations rgidity of the laminated medium has the form

$$\overline{B}_\Sigma = B_{0\Sigma}(1 + j\eta_\Sigma) = \operatorname{Re}\overline{B}_\Sigma + j \operatorname{Im}\overline{B}_\Sigma = \operatorname{Re}\overline{B}_\Sigma \left(1 + j\frac{\operatorname{Im}\overline{B}_\Sigma}{\operatorname{Re}\overline{B}_\Sigma}\right), \qquad (3.11)$$

where η_Σ is the loss factor, characterizing attenuation of vibratory energy in the flexurally-oscillating laminated medium, which is, according to (3.11) equal to

$$\eta_\Sigma = \frac{\operatorname{Im}\overline{B}_\Sigma}{\operatorname{Re}\overline{B}_\Sigma}. \qquad (3.12)$$

Thus, η_Σ can be found if the expression for complex flexural rigidity is known.

Flexural rigidity of a laminated medium can be determined by use of the bending moment in force in its section [99]

$$M_\Sigma = \overline{B}_\Sigma \theta_1' = \sum_{i=1}^{n} M_i + \sum_{i=1}^{n} N_i h_{i0}, \qquad (3.13)$$

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where $_1$ is sectional deflection of the first layer; M_i is the bending moment in a section of layer i; N_i is the tensile force of layer i applied in its neutral plane; $h_{i0}=h_{i1}-h_{10}$ is the force arm of N_i relative to the neutral plane of the laminated medium; h_{i1} is the distance between the neutral planes of layer 1 and layer i; h_{10} is displacement of the neutral plane of layer 1 when the remaining layers are joined to it.

The listed dimensions are shown in Fig. 11. It also depicts an element of the medium with length dx with indication of deformation which takes place within it.

The bending moment in a section of layer i is equal to:

$$M_i=B_i\theta_i', \tag{3.14}$$

where θ_i is the sectional deflection of layer i.

The force streching layer i when it is lengthened by quantity y_i (Fig. 11, b), is equal to

$$N_i=D_iy_i'=D_i\left[\sum_{j=1}^{i}\theta_j'h_j-\frac{1}{2}(\theta_1'h_1+\theta_i'h_i)-\theta_i'h_{10}\right]. \tag{3.15}$$

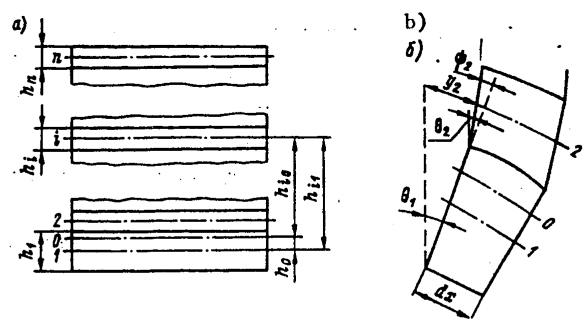

Fig. 11. Geometry (a) and deformation (b) of a system of n layers.

Key: 1, 2, i, n, neutral planes of the separate layers; 0 neutral plane of the system.

Inserting (3.14) and (3.15) into expression (3.13) we find that

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$$\overline{B}_{\Sigma}=\sum_{i=1}^{n}\left\{B_i\varphi_i'-D_ih_{i0}\left[\sum_{j=1}^{i}\varphi_j'h_j-\frac{1}{2}(\varphi_1'h_1+\varphi_i'h_i)-\varphi_1'h_{10}\right]\right\}, \quad (3.16)$$

where $\varphi_i'=\frac{\theta_i'}{\theta_1'}$; $\varphi_1'=1$.

In expression (3.16) $\phi 1'=\phi n'$, since the external surface of layer n is free and no shear deformation takes place in it. h_{10} and ϕ_i' (i=2, ..., n-1). To find h_{10} we use the equation to zero of the total tensile force in a section of the laminated medium

$$\sum_{i=1}^{n} N_i=0. \quad (3.17)$$

Inserting expression (3.15) into formula (3.17) we find that

$$h_{10}=\frac{\sum_{i=1}^{n} D_i\left[\frac{1}{2}(h_1+\varphi_i'h_i)-\sum_{j=1}^{i}\varphi_j'h_i\right]}{\sum_{i=1}^{n} D_i}. \quad (3.18)$$

The expression for ϕ_i' can be determined as follows. Shear deformations of internal layers ψ_i (i=2, ...; n-1) are formed due an increase in tensile force dN_{i+1}, exerted on an element of layer with length dx, equal to

$$dN_{i+1}=-\frac{\partial N_{i+1}}{\partial x}dx. \quad (3.19)$$

Force dN_{i+1} causes deformations ψ_i and ψ_{i+2} in adjacent layers, equal to

$$\psi_i=\frac{\zeta_{i+1}}{h_i}=\frac{\Delta N_i}{G_i dx};\quad \psi_{i+2}=\frac{\zeta_{i+1}}{h_{i+2}}=\frac{\Delta N_{i+2}}{G_{i+2}dx}, \quad (3.20)$$

where ζ_{i+1} is displacement of element i+1, caused by force dN_{i+1}; G_i is modulus of shear of layer i. Force dN_{i+1} is distributed between layers i and i+2, therefore, $\Delta N_i+\Delta N_{i+2}=dN_{i+1}$.

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If we equate values ζ_{i+1} in formulas (3.20) we get

$$\Delta N_i = \frac{dN_i}{1 + \frac{h_i G_{i+2}}{h_{i+2} G_i}}. \qquad (3.21)$$

Therewith, taking (3.19) into account, we have

$$\psi_i = \frac{\Delta N_i}{G_i dx} = -\frac{1}{G_i + G_{i+2}\frac{h_i}{h_{i+2}}} \frac{\partial N_{i+1}}{\partial x}. \qquad (3.22)$$

Taking into account that $\psi_i = \theta_1 - \theta_i$, and incorporating formula (3.15) and equation $\theta_i'' = -k_H{}^2\theta_i$ (k_H is the wave number of flexural oscillations in the laminated medium), from expression (3.22) we find that

$$\varphi_i' = \frac{\theta_i'}{\theta_i} = 1 - \frac{D_i k_H^2}{G_i + G_{i+2}\frac{h_i}{h_{i+2}}} \times$$

$$\times \left[\sum_{j=1}^{i} h_j \varphi_j' - \frac{1}{2}(h_1 + h_i \varphi_i') - h_{10} \right]. \qquad (3.23)$$

Inserting $B_i = B_{0i}(1 + j\eta_i)$ and $D_i = D_{0i}(1 + j\eta_i)$ into formulas (3.16), (3.18) and (3.23) and solving them simultaneously, one can determine the real and imaginary parts of the flexural rigidity of a system with any number of layers.

The complex flexural rigidity method for a three-layered medium is set forth in work [93]. A general case form this method is presented here. The results obtained in [99] represent a particular case of the expressions presented above.

The Wave Resistance Method. In a number of cases the displacement of the neutral plane of a plate, when an arbitrary set of layers is built up on it, is small in comparison with the plate thickness. Then one can use a method based on summation of wave resistance of the separate layers. As in the preceeding case, the requirement that the thickness of the separate layers be less than the length of the elastic waves in the system remains valid.

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The wave resistance of a uniform plate under longitudinal oscillations is [34]:

$$z_\zeta = j\omega m + \frac{Dk_n^2}{j\omega} + \eta \frac{Dk_n^2}{\omega}, \qquad (3.24)$$

where η is the loss factor of the plate.

Under flexural oscillations of a uniform plate its wave resistance is equal to [34]:

$$z_\xi = j\omega m + \frac{Bk_u^4}{j\omega} + \eta \frac{Bk_u^4}{\omega}. \qquad (3.25)$$

Layer applied to a damped plate introduce additional resistance to streching or bending. In the case of bending of a damped plate the applied layer, in addition, exerts resistance to longitudinal displacement of the plate's surface. This resistance is equal to [34]:

$$z_{\xi\zeta} = z_\zeta \left(\frac{k_u h_{12}}{2}\right)^2, \qquad (3.26)$$

where z_ζ is determined by formula (3.24); $k_И$ is the wave number of the damped plate; h_{12} is the distance between neutral susrfaces of the plate and the applied layer, equal to

$$h_{12} = \frac{1}{2}(h_1 + h_2);$$

h_1 and h_2 are thickness of plate and layer.

When n-1 layers are applied to flexurally-oscillating damped plate i the layer exerts resistance to longitudinal displacements of the plate's surface, equal to

$$z_{\xi\zeta i} = z_{\zeta 2i} \left(\frac{k_u h_{1i}}{2}\right)^2, \qquad (3.27)$$

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where

$$z_{\zeta 2i}=\frac{z_{\zeta 3i}+a_2}{1+z_{\zeta 3i}b_2};$$
$$z_{\zeta 3i}=\frac{z_{\zeta 4i}+a_3}{1+z_{\zeta 4i}b_3};\ \ldots;$$
$$z_{\zeta i-1,\,i}=\frac{z_{\zeta ii}+a_{i-1}}{1+z_{\zeta ii}b_{i-1}};$$
$$a_i=j\omega m_i;$$
$$b_i=\frac{\omega h_i}{G_i(1+j\eta_i)},$$

where $z_{ii}=z_i$ is determined by formula (3.24); h_{1i} is the distance between the neutral surfaces of the first and i layers.

The total wave resistance of a laminated medium consisting of n layers, including the damped plate, under longitudinal oscillations is equal to

$$z_{\zeta\Sigma}=\sum_{i=1}^{n}z_{\zeta i}, \tag{3.28}$$

where $z_{\zeta i}$ is determined by formula (3.24).

Under flexural oscillations of a laminated medium its total wave resistance is

$$z_{\xi\Sigma}=\sum_{i=1}^{n}z_{\xi i}+\sum_{i=2}^{n}z_{\xi\zeta i}, \tag{3.29}$$

where $z\xi_i$ is determined by formula (3.25); $z\xi\xi_i$ by formula (3.27).

The loss factor of a laminated medium is calculated by formula

$$\eta_\Sigma=\frac{\operatorname{Re} z_\Sigma}{|z_\Sigma|}. \tag{3.30}$$

Here z_Σ is determined by expression (3.28) or (3.29) depending on the type of oscillations in the laminated medium, $|z_\Sigma|$ is the modulus of the elastic or inertial part of z_Σ.

Determination of the loss factor of a laminated medium by the wave resistance method is done in works [34, 81] for a number of layers not greater than four. General expressions for an arbitrary number of layers are presented here. Results of the referenced works represent particular cases derived here.

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§10. Rigid Vibroabsorptive Coatings

The rigid vibroabsorptive coatings created by G. Oberst [94] and in our country by B.D. Tartakovskiy [28] consitute layers of rigid plastic, applied to the structure to be damped. The structure of this coating and the character of its deformation with flexure of the damped plate are shown in Fig. 12. It can be seen that deformation of the coating has a streching (contraction) character along its plane. The loss factor of a flexurally-oscillating plate, faced with a rigid vibroabsorptive coating, can be determined approximately be the wave resistance method (see §9.).

$$\eta = \frac{\eta_2 \alpha_2 \beta_2 (\alpha_2^2 + 12\alpha_{21}^2)}{1 + \alpha_2 \beta_2 (\alpha_2^2 + 12\alpha_{21}^2)}, \quad (3.31)$$

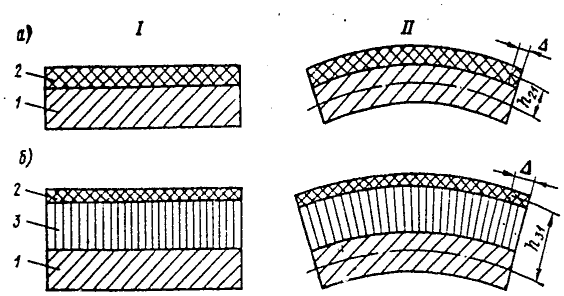

Fig. 12. Structure of a rigid vibroabsorptive coating (I) and the character of its deformation (II): a- rigid coating; b- rigid coating with intermediate layer.

Key: 1. plate being damped; 2. vibroabsorptive material (rigid plastic); 3. intermediate layer of rigid material; Δ- deformation of the vibroabsorptive material.

where η_2 is the loss factor of the material;

$$\alpha_2 = \frac{h_2}{h_1}; \quad \beta_2 = \frac{E_2}{E_1}; \quad \alpha_{21} = \frac{h_{21}}{h_1} = \frac{1+\alpha_2}{2};$$

h_1, h_2 are thickness of the damped plate and the layer of coating; E_1, E_2 are Young's modulus of the plate and coating; h_{21} is the distance between the neutral planes of the plate and the layer of coating.

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Formula (3.31) holds true under the condition that $\beta_2<10^{-2}$, which is practically always met.

The first term of expression (3.31) defines absorption of energy in a layer of plastic owing to its flexure and the second that owing to its streching. When $\alpha_2\beta_2(\alpha_2^2+12\alpha_{21}^2)<1$ coefficient $\eta\approx\eta_2\alpha_2\beta_2(\alpha_2^2+12\alpha_{21}^2)$. It can be seen that the loss factor of a plate with a rigid coating becomes less significant the greater the product of $\eta_2\beta_2$ or $\eta_2 E_2$, which called the mudulus of losses.

Fig. 13 shows the dependence of losses in a plate with rigid coating on the ratio of the thickness of coating and plate α_2, constructed for various β_2 by using formula (3.31). As this ratio increases the loss factor η increases, asymptotically approximating the value of the loss factor of the coating material η_2. With further increase in α_2 the increase in η ceases. This is explained by displacement of the neutral plane of the built-up plate toward the coating and accordingly by a decrease in strech deformation of the latter and absorption of energy within it. In practice they are limited to values on the order of 1.5-2. However, as shown in $20, these values are not always optimum.

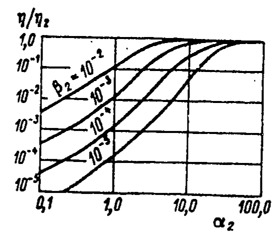

Fig. 13. Dependence of the loss factor of a plate η, faced with a rigid vibroabsorptive coating, on the ratio of thicknesses of coating and plate $\alpha_2=h_2/h_1$ at various ratios of $\beta_2=E_2/E_1$.

As a consequence of the nonlinear dependence of η on $\alpha_2=h_2/h_1$ the greatest value η for a given mass of vibroabsorptive material can be derived by applying it one side of the plate being damped. If an intermediate layer of light and rigid material is placed between the layer of rigid plastic and the plate being damped [27] then, as a result of moving the plastic layer away from the neutral plane of the plate, the strech deformation of the plastic increases and the loss factor in the structure increases accordingly. The loss factor of a plate with such a coating can be determined by a formula derived in the same way as formula (3.31):

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$$\eta \approx \frac{\eta_2}{1+\frac{a+12\alpha_3\alpha_{21}^2\beta_3 g_2\left(1+\eta_3^2\right)}{\alpha_3\beta_3\left[a\alpha_3^2+12\alpha_{31}^2 g_2^2\right]}}, \qquad (3.32)$$

where $a=(1+g_2)^2+\eta_3^2;\quad g_2=\frac{G_2}{E_3h_3k_и^2h_2};$

$$\alpha_3=\frac{h_3}{h_1};\quad \alpha_{31}=\frac{h_{31}}{h_1};\quad h_{31}=\frac{1}{2}(h_1+h_3)+h_2;\quad \beta_3=\frac{E_3}{E_1}.$$

Formula (3.32) holds true under the condition that $\alpha_2\beta_2(\alpha_2^2+12\alpha_{21}^2)\ll 1$, which is practically always met. Analysis of formula (3.32) shows [34] that η increases with rise in g_2. However, the increase in η practically ceases when parameter g_2 reaches a value equal to 10. Therefore, the minimum value of G_2 must be

$$G_{2\,\min}=10E_3h_3h_1k_и^2, \qquad (3.33)$$

Where E_3, h_3 are Young's modulus and thickness of the plastic layer; $k_И$ is the wave number of flexural oscillations of the damped plate with coating.

Exceeding the frequency at which condition (3.33) is met leads to a decrease in the loss factor of the plate with coating as a result of an increase in shear deformation of the intermediate layer and the corresponding decrease in strech deformation of the layer of vibroabsorptive material. This frequency equates to:

$$f_0=\frac{G_2c_{пр\,1}}{218E_3h_3}. \qquad (3.34)$$

In practice an increase in f_0 can be achieved at a given G_2 by a decrease in the tensile rigidity of the plastic E_3h_3, which will have a negative effect on the loss factor of the structure. Specifically, for a vibroabsorptive material with $E_3=2\cdot10^{10}$ DIN/cm^{2}and $h_3=0.2$ cm^2 ("Agat" plastic) applied to an intermediate layer of a material with $G_2=4\cdot10^8$ DIN/cm (foam-plastic PCV-1), $f_0=240$ hz.

The loss factor of a longitudinally-oscillating plate with a rigid vibroabsorptive coating can be derived by the wave resistance method

$$\eta=\frac{\eta_2}{1+\frac{1}{\alpha_2\beta_2}}\approx\eta_2\alpha_2\beta_2. \qquad (3.35)$$

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Since $\alpha_2\beta_2 \ll 1$, then relative to the longitudinal waves in the damped plate the rigid coating is less effective in comparison to absorption of energy of flexural waves. The same also holds true for a rigid coating with an intermediate layer in which, even without this, the effectiveness relative to longitudinal waves will be lessened by shear deformation of the intermediate layer. Special materials have been developed for rigid vibroabsorptive coatings on the basis of polyvinylchloride, polyvinylacetate and other polymers, as well as on the basis of epoxy resins. This is a number of patents on the chemical compositions of the indicated materials [12, 64, 80]. To give these materials high dissipative properties, the are impregnated with additives in the form of graphite, mica, vermiculite and other such substances. The physical and mechanical properties of the indicated materials are to a great degree dependent on temperature. Fig. 14 shows the dependence of the modulus of losses ηE of the vibroabsorbtive material "Antivibrit-2" on temperature T. It can be seen that $(\eta E)_{max}$ achieves maximum value at T=+20°C. With deviation form this temperature in either direction the value of E decreases markedly. Usually a vibroabsorptive material is considered effective in the temperature realm where $\eta E > 0.5(\eta E)_{max}$. Specifically, for the "Antivibrit-2" material the working temperature realm ranges $\Delta T = 0 \div 35$°C. A ΔT value on the order of 40°C is generally characteristic for materials of this type known at present. This is expained by the fact that polymeric materials have high vibration absorbing properties in a comparatively narrow temperature realm (the vitrification realm), in which the material changes from a glass-like to a rubber-like state (see $2.)

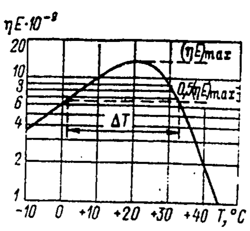

Fig. 14. Dependence of the modulus of losses ηE of the vibroabsorptive material "Antivibrit-2" on temperature T.

To expand the working temperature range of a rigid vibroabsorptive coating, it is recommended that two or more types of materials be used, the maximum effectiveness of which is at different temperatures [100]. These materials are applied to the structure being damped in layers. There is a rigid vibroabsorptive coating which incorporates an electric heating element, allowing regulation of the temperature of the material and the working temperature range of the coating [45]. So-called plasticizers are used to shift the working temperature range in the

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Table 2

Physio-Mechanical Properties of Rigid Vibroabsorptive Materials

Material	Country	Type of Material	Basis of Material	Temp of max effect T, °C	Loss Factor η	Young's Modulus $E \cdot 10^{-9}$, DIN/cm^2	Modulus of losses $\eta E \cdot 10^{-9}$ DIN/cm^2	Density ρ, g/cm
PCV Linoleum	USSR	Sheet	PVC	--	0.03	1.18	0.054	--
"Neva"	USSR	Mastic	--	--	0.016	4.0	0.064	--
BPM-1	USSR	Mastic	Bitumen	30	0.65	3.1	2.0	1.2-1.35
"Agat"	USSR	Sheet	PVC	20	0.25	20.0	5.0	1.35
"Antivibrit-2"	USSR	Mastic	EP	20	0.45	29.0	13.0	1.57
A-5	USSR	Mastic	EP	20	0.5	35.0	17.5	1.53
A-7	USSR	Mastic	EP	70	0.75	30.0	22.5	1.44
"Fon-Eks 62"	GDR	--	--	20	0.58	8.5	4.9	--
EIW-A3905	GDR	--	--	20	1.0	4.0	4.0	--
"Fonkiller 2023"	FRG	Mastic	PVA	20	0.26	40.0	10.4	1.22
"Shalshuk 163/91"	FRG	Mastic	PVA	20	0.25	17.0	4.2	0.6
LD-400	USA	Sheet	--	24	0.55	55.0	30.0	1.73
MRC-OG4	USA	Mastic	PVC/PVA	23	--	--	13.5	1.73

Notes: PVC - polyvinylchloride; PVA - Polyvinylacetate; EP - Epoxy Resin

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direction. An increase in the amount of plasticizer in the composition of the vibroabsorptive material lowers the high-effectiveness range of the material and vice versa.

The physio-mechanical properties of some materials, developed and used for rigid vibroabsorptive materials, are shown in Table 2, which was compiled from data in works [8, 9, 25, 34, 35, 56, 76, 83]. It is evident that our best domestic materials are not inferior to foreign materials in their vibration absorption properties; special materials are significantly more effective than materials used for finish work on ship structures (for example, compare PCV linoleum and "Agat" sheet material). Domestic vibration absorption materials are not inferior to foreign materials.

Fig. 15 presents frequency characteristics of loss factors of steel rods with layers of a rigid vibroabsorptive coating from the materials A-5 (USSR) and MRC-OG4 (USA) applied to them. Comparison shows that the effectiveness of these materials is high (η>0.1) and comparable.

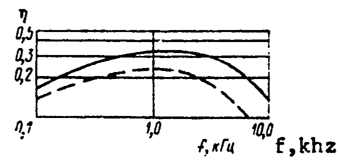

Fig. 15. Frequency characteristics of loss factors in steel rods with vibroabsorptive coatings applied to them.

Key: _____ A-5 material [25] ($\alpha_2=2$; $m_2/m_1=0{,}39$; $T=20°$ C)
----- MRC-OG4 material[56] ($\alpha_1=1{,}5$; $m_2/m_1=0{,}35$; $T=23°$ C)

Foam plastic of the PCV-1 type is usually used for intermediate layers. Its physio-mechanical properties are as follows: $G_2=4\times10^7$ DIN/cm^2, $\rho\sim0.1$, g/cm^3, $\eta\sim0.02$. From the point of view technology for applying coatings, the most advanced is the use of froth-forming polyurethane plastic foams for intermediate layers. Work [36] shows the possibility of using for this purpose PU-101 materials, which (after it hardens) has $E\sim10^9$ DIN/cm^2, $\rho=0.12$ g/cm^3.

The technology for application of a rigid coating depends on its properties. Sheet materials are applied with glue (Type PN-Eh or EhPK-519). The surface onto which the material is to be glued must be meticulously cleaned and primed. Special equipment for clamping the sheet plastic ensures a high-quality bond. Mastic materials are applied by dusting, spraying or stapling them on in layers 2-4mm thick until the required

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thickness is achieved. Mastic materials on the basis of polyvinylchloride and polyvinylacetate are sometimes made in the form of a water emulsion. After the water component dries out the layer takes on vibroabsorptive properties. Warming such materials is useful to accelerate the drying process. Materials based on epoxy polymers require a special heat treatment after application, without which they do not take on vibroabsorptive properties. The surface of mastic materials is machined after application to give them a decorative appearance.

§11. Stiffened Vibroabsorptive Coatings

A stiffen vibroabsorptive coating consitutes a layer of viscoelastic material on which a thin stiffening layer of rigid material is applied [77]. The structure of a stiffened coating and the character of its deformation under flexural oscillations of a damped plate are shown in Fig. 16.

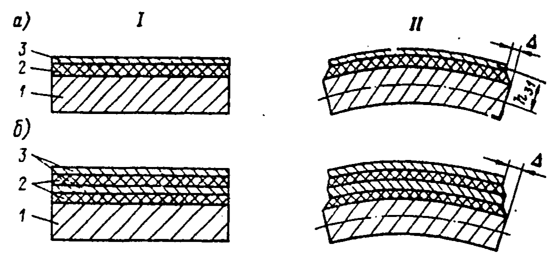

Fig. 16. Structure of a stiffened vibroabsorptive coating (I) an the character of its deformation (II): a - stiffened coating; b - multilayered stiffened coating

Key: 1. damped plate; 2. viscoelastic layer; 3. stiffening layer; Δ. deformation of the vibroabsorptive material

Absorption of vibratory energy in a stiffened coating is attributable to shear deformation in the viscoelastic layer. Rubbers or rubber-like plastics, i.e. pliable material, are usually used for this layer. Obviously these materials must also possess high internal loss properties. The loss factor of a plate faced with a stiffened vibroabsorptive coating is most simply determined by the wave resistance method (see §9).

$$\eta = \eta_2 \frac{\alpha_2^3\beta_2 + 12\alpha_{21}^2\alpha_2\beta_2 + 12\alpha_{31}^2 g_2\gamma_0 \left(\alpha_3\beta_3 - \mu_k^2\alpha_2\beta_2\right)}{1 + \alpha_2^3\beta_2 + \alpha_3^3\beta_3 + 12\alpha_{21}^2\alpha_2\beta_2 + 12\alpha_{31}^2\alpha_3 g_2\gamma_0 \left(1 + g_2 + \eta_2^2\right)}, \tag{3.36}$$

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where
$$\alpha_2=\frac{h_2}{h_1};\quad \alpha_3=\frac{h_3}{h_1};\quad \alpha_{21}=\frac{h_{21}}{h_1};\quad \alpha_{31}=\frac{h_{31}}{h_1};$$
$$h_{21}=\frac{1}{2}(h_1+h_2);\quad h_{31}=\frac{1}{2}(h_1+h_3)+h_2;\quad \beta_2=\frac{E_2}{E_1};$$
$$\beta_3=\frac{E_3}{E_1};\quad \gamma_0=\frac{1}{(1+g_2)^2+\eta_2^2 g_2^2};\qquad (3.37)$$
$$g_2=\frac{G_2}{E_3 h_3 k_н^2 h_2};\qquad \mu_k=\frac{k_{п2}}{k_н};$$

h_1, h_2, h_3 are thickness of the damped plate, the viscoelastic layer and the stiffening layer respectively; E_1, E_2, E_3 are the corresponding Young's modulus values; k_H $k_H \sim \eta\pi'_1$ is the wave number of the flexural oscillations of the plate with its coating. The first term of the expression (3.36) defines losses of energy attributable to flexure of the viscoelastic layer; the secon that attributable to streching; the third that attributable to shear. Insofar as the overall losses in the coating are caused by shear deformation, the first two terms of expression (3.36) may be disregarded ($\beta_2<\beta_3$). In addition, usually $\alpha_3\beta_3>\mu k^2\alpha_2\beta_2$. Taking the aforesaid into account, formula (3.36) takes on the form

$$\eta\approx\frac{\eta_2\gamma g_2}{(1+g_2^2)+g_2^2\eta_2^2+\gamma g_2\left[1+g_2\left(1+\eta_2^2\right)\right]},\qquad (3.38)$$

where

$$\gamma=\frac{12\alpha_{31}^2\alpha_3\beta_3}{1+\alpha_2^3\beta_2+\alpha_3^3\beta_3+12\alpha_{21}^2\alpha_2\beta_2}.\qquad (3.39)$$

Formula (3.36) corresponds to the more precise expression [99], derived by the complex flexural rigidity method, where $\alpha_3\beta_3\ll 1$ and $\alpha_{21}\alpha_2\beta_2\ll 2\alpha_{31}\alpha_3\beta_3 g_2$, which is practically always the case.

The dependence of the loss factor η on thickness of the coating layers and the damped plate is defined by the geometric parameter of the layer γ. The frequency dependence of η is defined by shear parameter , which, as can be seen from formula (3.37) is inversely proportional to frequency.

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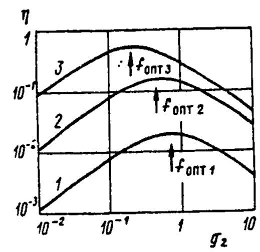

Fig. 17. Dependence of the loss factor of a plate, faced with a stiffened vibroabsorptive coating, on the shear parameter g_2, calculated at $\eta_2=1$. 1 — $\gamma=0{,}1$; 2 — $\gamma=1$; 3 — $\gamma=10$.

Figure 17 shows the dependence of the loss factor of a plate faced with a stiffened coating on the frequency-dependent parameter g_2, calcualted at $\eta_2=1$. It should be kept in mind that increase in g_2 corresponds to decrease in frequency. The optimum value $g_{2\,\text{ОПТ}}$ corresponds to frequency $f_{\text{ОПТ}}$, at which the loss factor of the plate has maximum value. With deviation from this frequency η decreases monotonically. It can also be seen from Fig. 17 that an increase in the geometric parameter γ leads to a rise in the loss factor of the plate η.

The optimum value of g can be easily derived by equating the derivative of η on g_2 to zero

$$g_{2\,\text{опт}}=\frac{1}{\sqrt{(1+\gamma)\left(1+\eta_2^2\right)}}. \tag{3.40}$$

Substituting formula (3.40) into expression (3.38) we find the maximum value η

$$\eta_{\max}=\frac{\eta_2\gamma}{\gamma+2\left(1+g_{2\,\text{опт}}\right)}. \tag{3.41}$$

The frequency, corresponding to $g_{2\,\text{ОПТ}}$, is equal to

$$f_{\text{опт}}=\frac{G_2}{2\pi E_3 h_3 h_2}\sqrt{\frac{E_1 h_1^3(1+\gamma)\left(1+\gamma_2^2\right)}{12 m_1}}, \tag{3.42}$$

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where m_1 is the mass of the plate, falling within one unit of surface.

Figure 18 shows the dependence of η_{max} on the geometric parameter γ at various values of η_2. The increase in η_{max} practically ceases when parameter γ reaches a value on the order of 10. It is not advantageous to increase the loss factor of a viscoelastic material η_2 more than one unit. It will be noted that the loss factor of a plate with a stiffened coating cannot be more than η_2; such a conclusion may be drawn form formula (3.38) by extending its parameter γ toward infinity. On the basis of this as well as formulas (3.41) and (3.42) one can construct a stiffened vibroabsorptive coating with maximum effectiveness placed at the frequency at which the greatest reduction in vibration of the structure to be damped is required. To increase losses in a damped plate, stiffened vibroabsorptive coating is sometimes applied in several layers.

From Fig. 17 it is evident that a stiffened vibroabsorptive coating is effective ($\eta > 0.7\ \eta_{max}$) in a frequency range encompassing approximately a decade (3.5 octaves). This range can be expanded if material, with specially selected frequency dependence in its physio-mechanical properties, is used for the viscoelastic layer. Work [17] shows that if material is used in which the shear modulus depends on frequency, as

$$G_2 = G_{02}\left(\frac{f}{f_0}\right)^{\alpha}, \qquad (3.43)$$

where G_{02} is the value of G_2 at frequency f_0, while α lies within the limits $1 > \alpha > 0$, the the band of frequencies in which the stiffened coating is effective is expanded. The dependence of the frequency band Δf, in which the stiffened vibroabsorptive coating is effective, on index α is shown below:

α	0	0.2	0.4	0.6	0.8	1.0
Δf, octaves	3.5	4.2	5.5	8.0	15.5	∞

Where $\alpha = 1$ the loss factor of the plate with a stiffened coating ceases to depend on frequency.

Relative to longitudinal oscillations of a damped plate, the stiffened coating, just as the rigid type, is little effective [34]. It is possible to create a stiffened coating with an intermediate layer [34]; however, due to technological difficulties this method is not in practical use.

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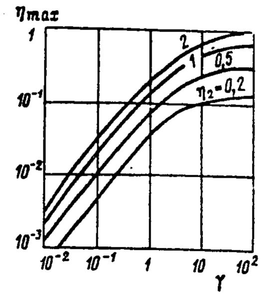

Fig. 18. Dependence of the maximum value of the loss factor of a plate, faced with a stiffened coating, on the geometric parameter γ.

Rubbers that offer oil resistance and other necessary operational qualities are used for the viscoelastic layer in stiffened coatings. The physio-mechanical properties of some of these rubbers are shown in Table 2. Sometimes an adhesive viscoelastic material is used, which at the same time serves as the binder. Stiffened coatings are known abroad which use such materials, called damping tape [74, 106]. Such coatings are convenient for damping circular structures since the technology for applying the coatins amounts to "bandaging." These materials are produced in rolls.

In domestic shipbuilding, the "Poliakril-V" stiffened vibroabsorptive coating is used for damping of structures made from light alloys [8 11]. As a viscoelastic layer it uses an acrylic polymer with $G=(2\div10)\times10^7$ DIN/cm^2 and $\eta=0.3\div0.5$. For comparison, we point to the American material of this type -- 3M-467, with $G=2\cdot10^7$ DIN/cm^2 and $\eta=0,6$[75, 76]. The thickness of stiffening layers in the "Poliakril-V" coating, which are made of aluminum foil, is 0.06mm and the thickness of the viscoelastic layer is 0.12mm. The number of layers N is determined depending on the thickness of the plate being damped h_1 by formula $N=1+h_1$, where h_1 is in mm. In this the relative mass of the coating ranges 40-50%. The loss factor of a plate faced with such a coating is shown in Fig. 19. The loss factor values of stiffened coatings made from materials MRC-0G4 and 3M-428A [56, 76] are also given for comparison. The effectivenesses of the compared domestic and foreign coatings, taking the difference in relative mass into account, are comparable.

The "Poliakril-V" coating is turned out on a rolling conveyor, after which it is held for 10-12 hours until polymerization of the adhesive viscoelastic layers is complete. The coating is joined to the structure being damped by the same viscoelastic layer. A reliable bond is formed after the coating is pressed to the structure (using cross-bar clamps, for example) for 48 hours. The coating can be applied to both primed and unprimed surfaces.

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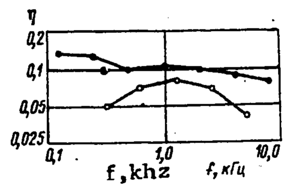

Fig. 19. Frequency characteristics of loss factors in plates faced with stiffened vibroabsorptive coating (T=20÷23°C).

Key: ● - "Poliakril-V" (m_{coat}/m_{plate}=0.4÷0.5 [8]
o - MRC-0G4 (m_{coat}/m_{plate}=0.25) [56]
● - 3M-428A (m_{coat}/m_{plate}=0.5 [76]

We will conclude this paragraph by pointing out some more complex stiffened vibroabsorptive coatings, which ensure high effectiveness with low relative mass. Work [57] describes a design which ensures intense shear deformation of the viscoelastic layer displaced from the neutral plane of the plate being damped by a significant distance by a special installation (Fig. 20,a).

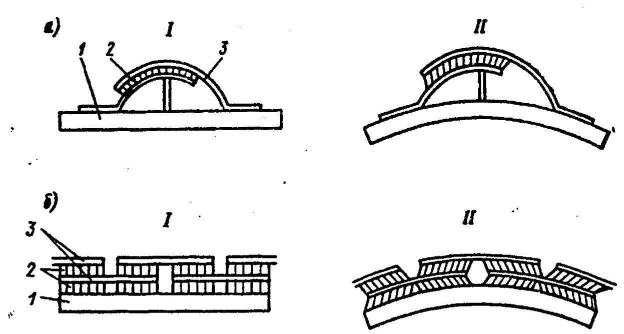

Fig. 20. Structure of stiffened vibroabsorptive coatings with increased effectiveness and character of their deformation under flexure of the damped plate: a - data from work [57]; b - data from [111]

Key: 1. damped plate; 2. viscoelastic layer; 3. stiffening element.

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Work [111] points out the possibility of achieving a loss factor of $\eta \sim 0.1$ with a relative mass of the layer of 0.03, in which the stiffening layer and the viscoelastic layer are split up into separate sectors which ensures greater shear deformations of the viscoelastic layers (Fig. 20,b).

§12. Pliable Vibroabsorptive Coatings

A pliable vibroabsorptive coating constitutes a layer of viscoelastic material (Fig. 21), in which elastic waves occur along the thickness of a flexurally-oscillating plate with lateral displacement of its surface [18, 114].

The length of the flexural wave in a damped plate at audio frequencies is much greater than the length of elasric waves in a viscoelastic layer; therefore, the elastic wave in such a layer can be imagined as a flat compression wave propagating normally toward the surface of the damped plate.

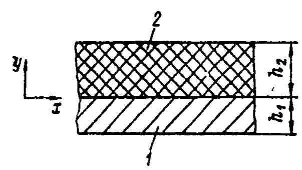

Fig. 21. Structure of a pliable vibroabsorptive coating

Key: 1. damped plate; 2. coating

The loss factor of a plate with a pliable coating, assuming that a flexural wave occurs in the plate and a compression wave occurs in the coating, is calculated by the deformation energy method in work [114]

$$\eta = \frac{\eta_2 [2\,\mathrm{sh}(\nu_2\eta_2) - \eta_2 \sin(2\nu_2)]}{2\mu_{12}\eta_2\nu_2 [\cos(2\nu_2) + \mathrm{ch}(\nu_2\eta_2)] + \eta_2 \sin(2\nu_2) + 2\,\mathrm{ch}(\nu_2\eta_2)}, \quad (3.44)$$

where η_2 is the loss factor of the coating material;

$$\nu_2 = k_2 h_2; \quad \mu_{12} = \frac{m_1}{m_2}; \quad m_1 = \rho_1 h_1; \quad m_2 = \rho_2 h_2;$$

k_2 is the wave number modulus of compression waves in the coating; h_1, h_2 are thicknesses of plate and coating; ρ_1, ρ_2 are densities of the plate and coating.

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Formula (3.44) holds true if $\eta_2{}^2<1$ and there is no load on the free surface of the coating. The typical dependence of the loss factor of a plate with a pliable vibroabsorptive coating is described by formula (3.44) and shown in Fig. 22. Analysis of this dependence shows that at low frequencies ($\nu_2<1$) expression (3.44) assumes the form

$$\eta \approx \frac{\eta_2 \nu_2^2}{3(1+\mu_{12})}. \qquad (3.45)$$

With decrease in frequency the loss factor η tends toward zero. At frequencies, where an uneven number of quarter waves falls along the thickness of the coating (spatial resonance), we have

$$\nu_{pn} = \left(n - \frac{1}{2}\right)\pi;$$

$$f_{pn} = \frac{2n-1}{4h_2} c_2 \quad (n=1,\ 2,\ 3,\ldots). \qquad (3.46)$$

Taking (3.46) into account, from expression (3.44) it follows that

$$\eta_{pn} = \frac{\eta_2}{1+\mu_{12}\eta_2\nu_{pn}\,\mathrm{th}\left(\frac{\nu_{pn}\eta_2}{2}\right)}. \qquad (3.47)$$

As can be seen from Fig. 22, the loss factor η at these frequencies assumes maximum values, with most of them occurring at the frequency of first resonance (n=1)

$$f_{p1} = \frac{c_2}{4h_2} \qquad (3.48)$$

and equal to

$(\mathrm{th}\,\nu_{p1}\eta_2/2 \to \nu_{p1}\eta_2/2)$

$$\eta_{p1} = \frac{\eta_2}{1+1{,}23\mu_{12}\eta_2^2}. \qquad (3.49)$$

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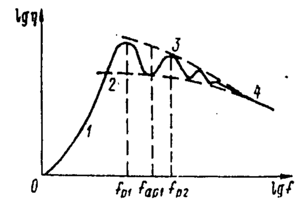

Fig. 22. Dependence of loss factor in a plate with pliable vibro-absorptive coating on freqency.

Key: 1. by formula (3.45); 2. by formula (3.51); 3. by formula (3.47); 4. by formula (3.52).

At frequencies of spatial resonance (a whole quantity of half-waves falls along the thickness of the coating)

$$\nu_{\text{ap}\,n} = \pi n;$$
$$f_{\text{ap}\,n} = \frac{nc_2}{2h_2} \quad (n = 1,\ 2,\ 3,\ \ldots), \tag{3.50}$$

consequently,

$$\eta_{\text{ap}\,n} = \frac{\eta_2}{1 + \mu_{12}\eta_2\nu_{\text{ap}\,n}\,\text{cth}\left(\frac{\nu_{\text{ap}\,n}\eta_2}{2}\right)}. \tag{3.51}$$

At antiresonant frequencies η has minimum values (see Fig. 22).

From formulas (3.47) and (3.51) it can been seen that with increase in frequency the loss factor approaches one and the same value, since when $\nu_2 > 2$ $\text{th}(\nu_2\eta_2/2) \approx \text{cth}(\nu_2\eta_2/2) \to 1$

$$\eta \approx \frac{\eta_2}{1 + \mu_{12}\eta_2\nu_2} \approx \frac{1}{\mu_2\nu_2} = \frac{\rho_2 c_2}{\omega m_1}. \tag{3.52}$$

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Thus at high frequencies the loss factor of a coated plate does not depend on the loss factor of the coating material. This corresponds to loading of a plate with a semi-infinite medium with acoustical resistance of $\rho_2 c_2$, in which energy radiated into the plate is totally absorbed.

As expression (3.48) shows, to lower the frequency of first resonance f_{p1} and extend the frequency spectrum, in which the pliable coating is effective, toward the lower frequencies, its thickness should be increased. However, such a possibility is limited due to the necessity to economize on displacement of ships.

Another way of lowering the frequency f_{p1} when h_2=const is to decrease the velocity of compression waves in the coating material c_2, equal to

$$c_2 = \sqrt{\frac{\lambda_2}{\rho_2}}, \tag{3.53}$$

where λ_2 is the elastic constant of the coating material for compression waves. From expression (3.53) it follows that to decrease c_2 it is necessary that ρ_2 be increased or λ_2 decreased. Increase in the density of the coating material ρ_2 while preserving λ_2=const can be achieved by adding particles of heavy metal to the viscoelastic material. Work [114] refers, for example, to a neoprene rubber with lead particles with ρ_2=3.3. To decrease the elastic constant λ_2 air cavities are made in the viscoelastic material [42]. The air content is selected in such a way that ρ_2 practically does not change.

Expression (3.52) can be rewritten as

$$\eta \approx \frac{\sqrt{\rho_2 \lambda_2}}{\omega m_1}, \tag{3.54}$$

from which it can been seen that a decrease of λ_2 (for the purpose of lowering f_{p1}) leads to degradation of losses in the damped plate at high frequencies. With an increase of the density ρ_2 for the same purpose the indicated losses increase. Thus, with with a given coating thickness h_2, using viscoelastic materials with metallic particles is the preferred way to lower frequency f_{p1}.

The loss factor of a vibroabsorptive coating η_{p1} reaches its greatest value at the frequency of first spatial resonance f_{p1}. As is evident

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from formula (3.49), ηp_1 depends on the loss factor of the coating material η_2, with ηp_1 passing through maximum at

$$\eta_2 = \frac{0{,}9}{\sqrt{\mu_{12}}} = 0{,}9\sqrt{\frac{m_2}{m_1}}. \qquad (3.55)$$

Substituting (3.55) into (3.49) we get

$$\eta_{p\,1\,\max} = 0{,}45\sqrt{\frac{m_2}{m_1}}. \qquad (3.56)$$

There is no sense in increasing η_2 higher than the limit of (3.55). The mass of the coating must be increased to derive a greter loss factor in the damped plate.

An analogy of the oscillatory behavior of a pliable coating is a rod with a length h_2, one end of which is excited by a longitudinal force. The resonant frequencies of the longitudinally-oscillating rod can be lowered by loading its free end with inertial resistance. Therefore, frequency fp_1 can be lowered while preserving the loss factor values at high frequencies by loading the free surface of the coating with metallic plating.

Distribution of displacements along the thickness of the coating in this case is expressed by the relationship

$$\zeta(y) = \zeta_0 \frac{\cos[k_2(h_2 - y)] + \alpha \sin[k_2(h_2 - y)]}{\cos(k_2h_2) + \alpha \sin(k_2h_2)}, \qquad (3.57)$$

where ζ_0 is amplitude of displacement of the surface of the damped plate at y=0;

$$\alpha = -j\,\frac{\omega z_H}{\lambda_2 k_2}; \qquad (3.58)$$

z_H is the load resistance per unit of area of the free surface of the coating at $y=h_2$.

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Metal platings are selected in such a way that their largest dimension l is much less that the length of the flexural wave in the damped plate ($k_H l<1$).

Since in this case the platings will not undergo flexural deformation, their resistance will be purely inertial and equal to $z_H = j\omega m_3$ (m_3 is the mass of the plating falling within one unit of area of the coating). Substituting expression (3.57) into formulas (3.5) and (3.9), for the loss factor of a plate with pliable coating and plating we get at low frequencies

$$\eta=\eta_2\frac{\nu_2^2\left(\frac{1}{3}+\mu_{32}+\mu_{32}^2\right)}{1+\mu_{12}}, \qquad (3.59)$$

where $\mu_{32}=m_3/m_2$. It will be noted that when $m_3=0$ expression (3.59) changes to (3.45). Comparison of these expressions shows that the presence of platings shifts the curve of the low-frequency asymptote of the loss factor of a plate with pliablecoating into the low frequency range in the ratio

$$\mu_f=\frac{f\,(m_3=0)}{f\,(m_3\neq 0)}=\sqrt{1+3\,\mu_{32}+3\mu_{32}^2}, \qquad (3.60)$$

where f is the frequency of the given loss factor value η.

The disposition of resonances and antiresonances of a pliable coating with platings can be determined from the expression for its resistance relative to normal forces from the direction of the plate

$$z_2=j\rho_2 c_2\frac{\alpha\cos\nu_2+\sin\nu_2}{\cos\nu_2-\alpha\sin\nu_2}. \qquad (3.61)$$

At spatial resonance $z_2\to\infty$, since compression waves that are reflected from the surface $y=h_2$ arrive at the excited surface ($y=0$) in antiphase to its oscillations. Consequently, the condition of resonance is

$$\cos\nu_2=\alpha\sin\nu_2, \qquad (3.62)$$

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or $$\nu_2 \operatorname{tg} \nu_2 = \frac{1}{\mu_{32}}. \quad (3.63)$$

Under an antiresonant condition, when the reflected waves are in phase with oscillations of the surface of the plate at y=0, z2 will tend toward zero, consequently

$$\sin \nu_2 = -\alpha \cos \nu_2, \quad (3.64)$$

or $$\frac{\operatorname{tg} \nu_2}{\nu_2} = -\mu_{32}. \quad (3.65)$$

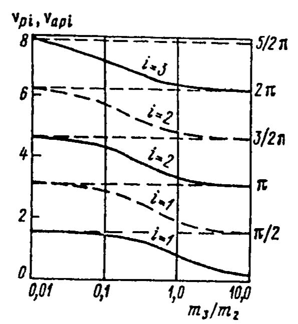

Fig. 23. Dependence of resonant ν_{pi} and and antiresonant ν_{api} values of parameter ν on m3/m2.

Figure 23 shows values ν_2, at which resonances and antiresonances are observed, depending on μ_{32}. A mass of the plating, commensurate with the mass of the viscoelastic layer of the coating, is required for an appreciable decrease in the frequency of first resonance. Values of the loss factor of the damped plate at resonant and antiresonant frequencies, when platings are present, will be defined by formulas (3.47) and (3.51) with corresponding values of parameter ν_2, determined from the graphs in Fig. 23. The loss factor of a coating with platings at high frequencies is determined by formula (3.52). Figure 24 shows the frequency characteristics of the loss factor of a steel plate 6mm thick, faced with a pliable vibroabsorptive coating of rubber with air cavities, ans steel platings 5.0x5.0x0.4 mm^3 (μ_{32}=2.65) and without them. Use of the platings does not alter the loss factor of a damped plate at high frequencies and expands the frequency spectrum, in which the coating is effective, into the lower frequencies in the ratio μ_f=6.2. Calculation of this ratio by formula (3.60) gives the value μ_f=5.5, which agrees well with the experiment.

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Table 3

Dynamic Modulus of Shear G and Loss Factor η of Rubbers

Sort of Rubber	$G \cdot 10^{-8}$, DIN/cm^2	η
1002	1.0	0.6
1011	1.2	0.2
5569	1.2	0.4
278-4	0.6	0.27
922	0.3	0.35
615	0.18	0.27

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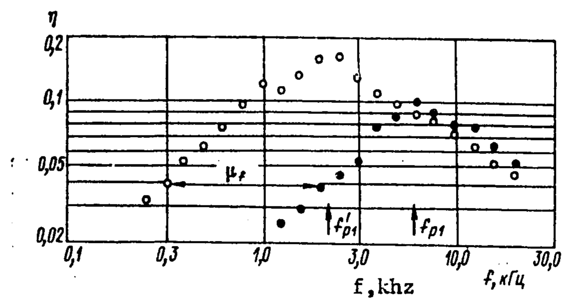

Fig. 24. Loss factor of a pliable vibroabsorptive coating.

Key: O - with metallic platings
● - without platings

With application of a pliable vibroabsorptive coating to a longitudinally-oscillating olate, flat shear waves will be excited it, which propagate along its thickness h2. The loss factor in such a plate will be defined by formula (3.44) when k2 is replaced by k_c2. Since the shear modulus of the viscoelastic material G_2 is less than its elastic constant for compression waves λ_2, the range in which the pliable vibroabsorptive coating is effective relative to longitudinal oscillations of the damped plate will be low-frequency. At high frequencies the loss factor in a longitudinally-oscillating plate will be less that in a flexurally-oscillating plate, since $c_{c2}<c_2$[see formula (3.52)].

Several grades of sheet rubber produced by domestic industry can be used for manuafacture of pliable vibroabsorptive coatings (Table 3). Air cavities can be made in this rubber either by the use of a special press of by gluing together narrow strips of rubber with a gap between them. The ratio of the air occlusions in the rubber, sufficient to give the mass the required elastic properties, must be on the order of 0.1-0.2. Domestic industry does not produce rubber materials with metal additives.

§13. Combination Vibroabsorptive Coatings

Combination vibroabsorptive coatings comprise several of the mechanisms for absorption of vibratory energy examined in §10-12. Owing to such a combination, they can either expand the frequency spectrum in which the coating is effective or increase its loss factor at a given frequency.

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The loss factor of a combination coating is determined by the formula:

$$\eta_{\Sigma}=\sum_i \eta_i, \qquad (3.66)$$

where $_i$ is the loss factor attributable to i mechanism for absorbing vibratory energy, calculated, depending on the type of absorption mechanism, by formulas (3.31), (3.32), (3.38) and (3.44).

Some of the possible designs for combination coatings are examined below (Fig. 25).

Pliable-Stiffened Vibroabsorptive Coating. In this coating, which consists of a pliable viscoelastic layer and a stiffening sheet, the mechanisms for losses attributable to shear and thickness deformations of a viscoelastic layer are combined. The structure and frequency dependence of components of the total loss factor are shown in Fig, 25,b. Factors η_C and η_T are calculated in accordance with formulas (3.38) and (3.44). In calculation of η_T it should be kept in mind that the stiffening layer constitutes a sheathing for the coating which has z_3=jωm3. The ratio of frequencies f_C and f_T, at which maximum η_C and η_T are observed, in accordance with formulas (3.42) and (3.48), is

$$\mu_f=\frac{f_T}{f_c}\approx\frac{\pi}{2}\sqrt{\frac{E_3\rho_1h_1h_3}{G_2\rho_2h_2^2}}=\frac{\pi}{2}\frac{c_{n3}}{c_{c3}}\sqrt{\frac{m_1m_3}{m_2^2}}. \qquad (3.67)$$

Here and below the indices of the layers are listed beginning with the plate being damped. In this parameter γ of the stiffened coating is assumed to be substantially greater than one. For the greatest possible expansion of the coating's frequency spectrum it is necessary to select parameters of the coating's layers in such a way that μ_f is on the order of 100.

The combination coating being examined was studied experimentally. The loss factor was measured for designs consisting of a steel plate to be damped 0.6mm thick, a layer of rubber 1.2cm thick (ρ_2=1 g/cm^3, G_2=10^8 DIN/cm^2, η_2=0.6) containing 15% air cavities by volume, (c_2=4·10^4 cm/c), and a stiffening sheet of glass-plastic (h_3=0.25 cm, ρ_3=1.7 g/cm^3, E_3=10^{11} DIN/cm^2). Results of the measurements and calculation by formulas shown in Fig. 26 correlate well.

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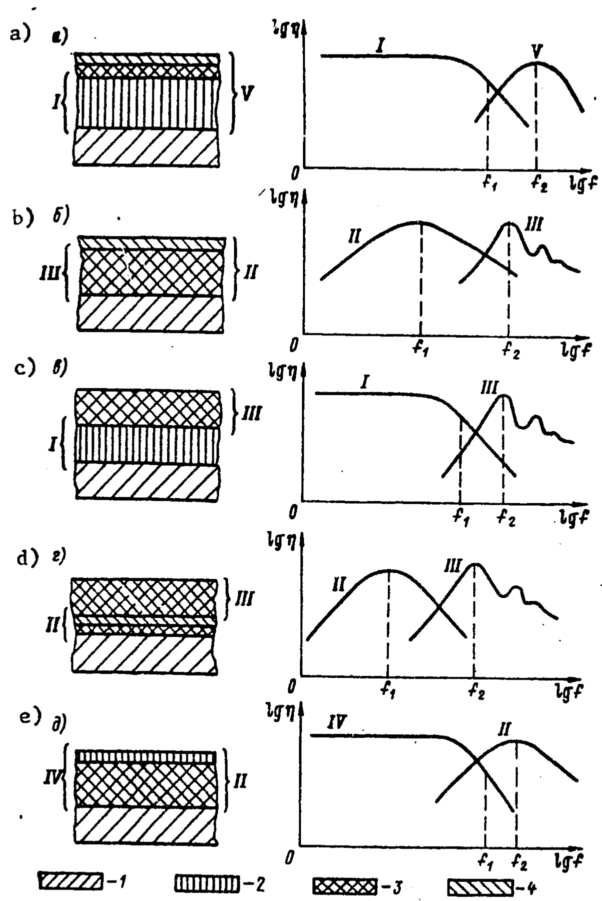

Fig. 25. Structure and frequency characteristics of the loss factor in combination vibroabsorptive coatings: a) rigid-stiffened (with intermediate layer); b) stiffened-pliable; c) rigid-pliable; d) pliable-stiffened; e) rigid (with intermediate layer) - stiffened.

Key: I- rigid coating; II- stiffened coating; III- pliable coating; IV- rigid coating with intermediate layer; V- stiffened coating with intermediate layer.
1. damped plate; 2. rigid viscoelastic layer; 3. pliable viscoelastic layer; 4. stiffening layer.

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Rigid (with intermediate layer) -Stiffened Vibroabsorptive Coating. A rigid coating with an intermediate layer work effectively until, with increase in frequency, shear deformations do not occur in the intermediate layer and it ceases to transmit streching forces to the viscoelastic layer when the damped plate is flexed. Such deformations occur when 1/6 and more of the length of the shear wave falls along the thickness of the intermediate layer, i.e. condition $k_c 2h2>1$ will be met.

In accordance with this condition, the frequency, beginning at which the loss factor of a rigid coating with intermediate layer η_p starts to drop substantially, will be equal to

$$f_1 = \frac{c_{c2}}{2\pi h_2}. \quad (3.68)$$

Effectiveness of the coating can be improved if a material, which offers vibratory energy losses under shear, is used for the intermediate layer [2]. In this case at the appropriate frequency such a coating would work as a stiffened coating, the loss factor of which achieves maximum value at frequency f_2, which is determined by formula (3.42). Structure and and frequency characteristics of components of the total loss factor of the coating are shown in Fig. 25,d. Loss factors η_p and η_c attributable to streching of the rigid viscoelastic layer and shear of the intermediate layer are calculated by formulas (3.32) and (3.38) respectively. The relationship of frequencies f_2 and f_1 are equal to ($\gamma>1$)

$$\mu_f = \frac{f_2}{f_1} \approx \sqrt{\frac{G_2 h_2^2 \rho_2}{E_3 h_3 h_1 \rho_1}} = \frac{c_{c2}}{c_{n3}} \sqrt{\frac{m_2^2}{m_1 m_3}}. \quad (3.69)$$

It is advisable to choose a value μf in this case as close as possible to one.

Rigid-Stiffened (with intermediate layer) Vibroabsorptive Coating. With increase in frequency the loss factor of a rigid coating decreases, when the thickness of the coating become comparable to the length of the shear wave. The condition under which the decrease occurs can be approximately presented as $k_c 2h_2>1$, from which the frequency, which defines the boundary of effective performance of the coating, is

$$f_1 = \frac{c_{c2}}{2\pi h_2}. \quad (3.70)$$

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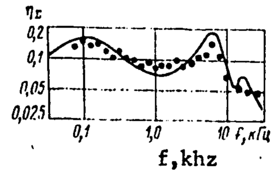

Fig. 26. Total loss factor of a combination pliable-stiffened vibroabsorptive coating.
—— calculation; ● - experiment.

If a stiffened coating is applied over the rigid coating then at certain frequencies such a combination coating will perform as a stiffened coating with an intermediate layer, which increases shear deformations of the viscoelastic layer [71, 91]. Structure and frequency characteristics of components of the total loss factor of the coating are shown in Fig. 25,a. The components η_p and η_c, attributable to streching of the rigid viscoelastic layer and shear of the pliable viscoelastic layer, are calculated by formulas in work [34] and (3.32). Maximum of the component η_c is observed at frequency f_2, determined by formula (3.42). The ratio of frequencies f_2 and f_1 is ($\gamma>1$)

$$\mu_f = \frac{f_2}{f_1} \approx \frac{h_2}{h_3} \sqrt{\frac{G_3^2 h_2^2 \rho_2}{E_4 G_2 h_1 h_4 \rho_1}} = \frac{c_{c3}^2 h_2^2}{c_{c2} c_{n4} h_3^2} \sqrt{\frac{m_3^2}{m_1 m_4}}. \quad (3.71)$$

Value μ_f should be chosen as close as possible to one. In this the frequency of the combination coating will be broader than that of a rigid coating.

Stiffened-Pliable Vibroabsorptive Coating. The load (stiffening) layer of a stiffened coating repeats the lateral displacements of a damped plate. Therefore, when a pliable coating is applied over the stiffened layer the effectiveness of the former will be the same as if it were applied directly to the plate being damped. With suitable selection of parameters for a such a combined coating, frequencies f_1 and f_2, at which the loss factors of the stiffened and pliable coatings have maximum value, can be widely spaced, thereby expanding the frequency spectrum in which the combination coating is effective.

Structure and frequency characteristics of the components of the total loss factor are shown in Fig. 25,d. Components of the loss factor η_c

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and η_T, attributable to shear of the inner viscoelastic layer and thickness deformation of the outer viscoelastic layer, are calcualted by formulas (3.38) and (3.44). The ratio of frequencies f_2 and f_1 is

$$\mu_f = \frac{f_2}{f_1} \approx \frac{\pi}{2}\sqrt{\frac{G_4 E_3 h_1 h_3 \rho_1}{G_2^2 h_4^2 \rho_4}} = \frac{\pi}{2}\,\frac{c_{c4} c_{n3} h_3}{c_{c2}^2 h_4}\sqrt{\frac{m_1 m_3}{m_2^2}}. \qquad (3.72)$$

Value μ_f, essential for substantial expansion of the frequency spectrum in which the coating is effective, must be on the order of 100.

Rigid-Pliabe Vibroabsorptive Coating. The outer surface of a rigid coating also repeats lateral displacements of the damped plate. Therefore, a pliable coating applied on top of a the rigid one will perform with the same effectiveness as when it is applied directly to the plate. Having suitably selected parameters for these coatings, effectiveness of the combined coating can be increased at frequencies exceeding value f_1, determined by formula (3.70).

Structure and frequency characteristics of components of the total loss factor of such a combination coating are shown in Fig. 25,c. Components η_p and η_T, attributable to stretching of the middle layer and thickness deformation of the outer layer, are calculated by formulas (3.31) and (3.44). The ratio of frequencies f_2 and f_1 is

$$\mu_f = \frac{f_2}{f_1} \approx \frac{\pi h_2}{2h_3}\sqrt{\frac{G_3\rho_2}{G_2\rho_3}} = \frac{\pi c_{c3} h_2}{2c_{c2} h_3}. \qquad (3.73)$$

It is advisable in this case to select a ratio μ_f near one.

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Chapter 4. VIBROABSORPTIVE CONSTRUCTION MATERIALS SUITABLE FOR USE ON SHIPS

§14. Laminated Vibroabsorptive Materials

Some materials which possess internal loss properties are suitable for manufacture of ship structures. Strucures of such materials do not require vibroabsorptive coatings which constitutes an indisputable advantage. Vibroabsorptive construction materials include so-called laminated vibroabsorptive materials, consisting of two metallic plates (usually of equal thickness) joined together by a viscoelastic adhesive layer. In foreign technical literature these are called "sandwiches." Such construction materials can be used in shipbuilding for sound insulating housings, light-duty bulkheads, [pyoles], walls of bunkers into which dry cargoes are poured and other elements of the hull-frame structure which do not bear significant static loads.

Structure and character of deformation of the viscoelastic layer under flexural oscillations of the laminated vibroabsorptive material are shown in Fig. 27. Here the viscoelastic material, just in stiffened vibroabsorptive coatings, undergoes shear deformation.

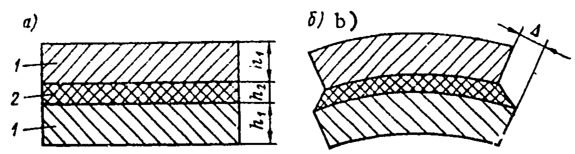

Fig. 27. Structure (a) and character of deformation of a laminated vibroabsorptive material (b).

Key: 1. metal plates; 2. viscoelastic material; Δ. deformation of the viscoelastic material.

The loss factor of a laminated vibroabsorptive material can be determined by the complex flexural rigidity method (see §9). For a symmetrical structure of such material ($h_1=h_3$ – thicknesses of the metal plates which make up the structure), which are often used in practice, the following expression for loss factor is derived in work [21]:

$$\eta = \frac{2\eta_2 g\gamma}{1+2g(2+\gamma)+4g^2(1+\gamma)}, \qquad (4.1)$$

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where η_2 is the loss factor of the viscoelastic material; G_2 is the modulus of shear of the viscoelastic material; h_1, h_2 are thicknesses of the metal and viscoelastic layers; E_1 is Young's modulus of the metal plate material; $k_И$ is the wave number of flexural oscillations of the structure, approximately equal (with slight error at low frequencies) to $k_{И\ пл1}$; $\gamma=3(\alpha_2+1)$ is the geometric parameter; $\alpha_2=h_2/h_1$.

Formula (4.1) holds true under the condition that E_1h_1 E_2h_2, which is practically always met. Analysis of formula (4.1) shows that dependence of the loss factor η on frequency reaches maximum when

$$g_{\text{опт}}=\frac{1}{2\sqrt{1+\gamma}}, \tag{4.2}$$

which is equal to

$$\eta_{\max}=\frac{\eta_2\gamma}{2\sqrt{1+\gamma}+2+\gamma}. \tag{4.3}$$

Figure 28 shows the dependence of ratio η_{max}/η_2 on the geometric parameter γ. Construction of this dependence takes into account that $\gamma>3$ ($\gamma=3$ at $h_2=0$). Increase in α_2 and γ leads to a rise in η_{max}. For structural reasons it is difficult to ensure $\alpha_2>1$. Therefore, the value of η_{max} may not exceed $(0.3\div0.5)\eta_2$. Consequently, achievement of high values of η_{max} requires viscoelastic materials with high internal losses ($\eta_2\sim1$). The optimum frequency corresponding to the value of the shear parameter described by formula (4.2) is

$$f_{\text{опт}}\approx\frac{G_2\sqrt{1+\gamma}}{\pi\sqrt{12}\,h_2\sqrt{E_1\rho_1}}. \tag{4.4}$$

Through formulas (4.3) and (4.4) parameters of the laminated vibro-absorptive material can be selected so that its maximum effect is at the frequency where the greatest reduction in vibration is needed. Usually this frequency is taken to be 1.0 khz.

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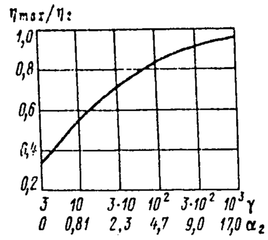

Fig. 28. Dependence of the maximum loss factor of a laminated vibroabsorptive material on the geometric parameter γ.

Laminated vibroabsorptive materials are developed at present which are suitable for use in shipbuilding. Work [21] describes such a material made from dural sheets 0.15cm thick and ML-25 mastic 0.1 cm thick. ML-25 mastic is modified bitumen with additives of graphite and epoxy resin to increase its adhesive properties. The loss factor of this material is on the order of 0.6 at a frequency near 1.0 khz (Fig. 29). ML-25 mastic has the following characteristics: $G_2=2\cdot10^8$ DIN/cm^2, $\eta_2=1.13$. Calculation of η_{max} by formula (4.3) gives a value of $\eta_{max}=0.47$, which agrees satisfactorily with experiemental data shown in Fig. 29.

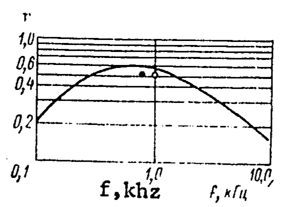

Fig. 29. Loss factor of laminated vibroabsorptive materials (T 20°C).

Key: — material on the basis of ML-25 mastic;
● "Viponit" material;
O "Bondall" material

Developed also is the laminated vibroabsorptive material "Viponit" from dural or steel sheets. The maximum loss factor of this material ($h_1=h_3=0.05\div0.3$ cm, $h_2=0.04$ 0.08 cm) is $\eta_{max}=0.5$ at a frequency about 1 khz at temperature +20°C. The mastic used in "Viponit" is manufactured on a polyvinylacetate base. This material can be welded, bent, cut and riveted [10].

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Glass-cloth bonded to metal sheets with epoxy glue can be used instead of glue and pliable plastic as a viscoelastic layer [54]. Such a design gives a low loss factor value, but its effectiveness is practically independent of temperature.

One of the foreign laminated vibroabsorptive materials is "Bondall", produced by the Hoch firm (FRG). The maximum loss factor of this material is ηmax 0.5 at a temperature of T=20°C. It can be molded, cut, welded and streched [79].

Work [115] refers to a laminated vibroabsorptive material consisting of two steel layers with a thermoplastic intermediate layer, joined together mechanically or by spot-welding. The material is produced in 2.5x1.25 m^2 sheets in thicknesses from 0.9 to 2.5 mm. Either stainless steel or common steel with a decorative coating is used in manufacture of this material. The material can be machined by usual methods. The use of this material instead of standard steel sheets reduces vibration by 12-20 db.

According to information from foreign technical literature, laminated vibroabsorptive materials are being widely used not only in shipbuilding, but in other spheres of industry [61, 102, 103, 104]. Using them for sound insulating housings for noise-generating machinery and equipment, vibrating transport chutes, loading funnels, cleaning drums, etc. gives an effective reduction of vibration and noise of 10-20 db.

§15. Vibroabsorptive Alloys

Some metals and alloys possess significant internal loss properties. These losses are particularly high in dual-phase materials, in which the basic structure (phase) contains a greater quantity of finely divided impurities than the second phase. Absorption of energy in such materials, when they undergo deformation, takes place primarily at the boundaries between phases. For instance, in cast iron the absorption of energy takes place at the boundary between the metal base and the particles of graphite. Internal losses in pure metals are usually insignificant. The internal loss factor of iron is on the order of $\eta\sim(2\ 6)\cdot10^{-4}$. At the same time the internal loss factor of cast iron is $\eta\sim10^{-2}$. Comparatively high internal losses occur in dual-phase alloys based on combinations of manganese-copper, nickel-titanium, and other alloys.

Manganese-copper alloys are those most widely used in vibroabsorptive construction materials. These alloys posses relatively high dissipative properties as well as good strength qualities and they can be worked hot or cold. High internal losses occur with the proper thermal treatment of an alloy with a certain ratio of components. Work [26]

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describes an alloy with a manganese content on the order of 60%, which is annealed at a temperature of 425°C for 1.5-2 hours. Internal losses in this alloy, as in the majority of others, depend on internal tensions which occur with oscillations. Its internal loss factor is on the order of 0.02 under flexural oscillations with amplitudes up to 0.1 mm of a specimen 2 mm thick and it increases at higher amplitudes. The high dissipative properties of the alloy are preserved when it is heated up to 50°C, after which they decline sharply; subsequent cooling of the alloys restores these properties.

Inasmuch as substantial absorption of vibratory energy in alloys of manganese and copper occurs at quite high amplitudes of oscillation, it is advisable that these alloys be used for parts and units in machinery which generates a high level of vibration. Work [4] presents results of acoustical tests of a VMN-5 pump, the body and frame of which was made of the domestic manganese-copper alloy "Aurora" [ρ=7.4 g/cm^3, $\eta=(1\div5)\cdot10^{-2}$]. Comparison of this machine's vibration with vibrations of a like machine made of traditional materials shows a reduction in vibration level on an average of 5 db over the frequency spectrum from 0.1 to 10.0 khz (Fig. 30).

Reduction in level of vibration, db

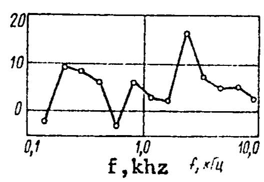

Fig. 30. Frequency characteristics of the reduction of vibration levels of a mechanism made from a manganese-copper vibroabsorptive alloy.

Vibroabsorptive alloys are used in foreign technology for manufacture of assemblies and foundations for machinery. Work [59] describes the manganese-copper alloy "Sonoston," which possesses a loss factor on the order of $\eta\sim0.07$. The same work points out the use of vibroabsorptive alloys of powdered iron or cast iron with a high graphite content ($\eta\sim0.02$). Developed and in use also is a vibroabsorptive alloy based on iron, which preserves high dissipative characteristics when heated up to 350°C [108]. The work points out the possibility of using in shipbuilding vibroabsorptive alloys of manganese and copper which are effective at temperatures of 100°C and higher. The alloys are not inferior in strength to steel and they ensure a noise reduction of 10 db.

More detailed information on the physio-mechanical properties of vibroabsorptive alloys can be found in work [52]. Work [84] presents information on loss factors of various metals and alloys.

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§ 16. Non-Metallic Vibroabsorptive Materials

Many of the non-metallic construction materials used in shipbuilding have definite, albeit insignificant in comparison with special materials, vibration absorbing capabilities. In a number of cases the use of these materials is preferable to the use of metallic materials. The use of non-metallic sheet material instead of metal sheets for sheathing of compartment walls can appreciably reduce such a defect as rattling which is caused by excitation in the sheating of resonant oscillations.

In choosing the type of construction material for sheathing or light walls, decorative trim, etc. from the point of view of their vibroabsorptive properties, a knowledge of their physio-mechanical characteristics is essntial: loss factor, Young's modulus and density (Table 4). This table was compiled on data from work [7]. The same work sets forth for comparison analogous characteristics of metals which are used for ship structures. It can be seen from the table that most non-metallic construction materials have loss factors several times greater than the loss factors of metal ship structures.

Attention should be directed to glass-plastic, which has certain technological advantages which allow light-duty bulkheads, superstructures and, in some cases, even the entire hull of a ship to be made from this material.

Loss factor values for most non-metallic materials listed in Table 4 depend little on frequency. As an example, Fig. 31 shows the frequency characteristics of the loss factor of a plate of glass-plastic 1.2 cm thick. In the 0.1-10.0 khz frequency spectrum the value of the loss factor changed only within the limits of $(1 \div 2)10^{-2}$.

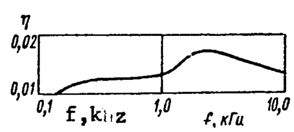

Fig. 31. Frequency characteristics of internal losses in glass-plastic.

In addition to elements of ship hull-frame structures, certain machine assemblies can be made from non-metallic materials which have substantial vibroabsorptive properties. The acoustical effect achieved thereby is very significant, particularyly at high frequencies. Work [50] shows results of measurements of the vibrations from three identical reduction gears, the housing and chassis of which were made

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of Silumin, glass-plastic and [voloknit]. The reduction of vibration levels of the second and third gears, in comparison with the first, was 21 db and 14 db respectively. Work [42] points out that installation of a glass-plastic housing on a internal combustion engine reduced the engine's vibration to 5 db at high frequencies. The use of a caprolon drive screw instead of a steel one in an oil pump lowered its vibration by 8-16 db in the 2.0-20.0 khz frequency spectrum [40].

Chapter 5. OTHER MEANS OF VIBRATION ABSORPTION

§ 17. Local Vibration Absorbers

The absorption of vibratory energy in plates of ship structures can be increased by installation of local vibration absorbers. The simplest of such vibration absorbers are rubber-metallic audio frequency antivibrators [19], which constitute a system with one degree of freedom with a dissipative element. A rubber layer is used for the element and it at the same time serves as the elastic element of the system. Possible designs for the antivibrators are suggested by I.I. Klyukin (Fig. 32).

With installation of an antivibrator in the antinode of a plate's oscillations, force is exerted on it which is directed along its axis. Displacements of the mass of the antivibrator under influence of this force cause compression deformations (Fig. 32, a,b) and shear deformations (Fig. 32,c) of the dissipative element.

The loss factor of a plate, in the antinode of oscillations of which the antivibrator is installed, is approximately equal to [34]:

$$\eta = \frac{4M_0\eta_0M_f^2}{M_{\text{пл}}\left[\eta_0^2 + (\mu_f^2 - 1)^2\right]}, \tag{5.1}$$

where M_0 is the mass of the antivibrator; η_0 is the loss factor of the rubber; $M_{\text{пл}}$ is mass of the plate.

$$\mu_f = \frac{f}{f_0};$$

f_0 is the resonant frequency of the antivibrator at $\eta_0=0$.

The resonant frequency of the antivibrator can be determined approximately by the formula:

-- for design shown in Fig. 32, a, b

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Table 4

Physio-Mechanical Characteristics of Non-Metallic and Metallic Construction Materials Used in Shipbuilding

Material	η	E, DIN cm	ρ, g/cm^3
Glass-plastic	$1-2\cdot10^{-2}$	$1-2\cdot10^{11}$	1.7
Plywood	$1.3\cdot10^{-2}$	$3.4\cdot10^{10}$	0.8
Pine boards	$1\cdot10^{-2}$	$1\cdot10^{11}$	0.5–0.8
Organic glass	$5\cdot10^{-2}$	$3.1\cdot10^{10}$	1.2
"Ramolit-1" wood-fiber sheets	$2\cdot10^{-2}$	$3.0\cdot10^{10}$	1.0
"Asbosilit 609" mineral-fiber	$1\cdot10^{-2}$	$3.0\cdot10^{10}$	0.8
FSM-1 glass textolite sheets	$1.7\cdot10^{-2}$	$5\cdot10^{10}$	1.3
PCV-1 foam-plastic sheets	$3.8\cdot10^{-2}$	$1\cdot10^{9}$	0.1
FS-7 foam-plastic sheets	$2.1\cdot10^{-2}$	$3.4\cdot10^{8}$	0.1
FF foam-plastic sheets	$3\cdot10^{-2}$	$5\cdot10^{8}$	0.16
Steel	$1\cdot10^{-4}$	$2.1\cdot10^{12}$	7.8
Duraluminum	$5\cdot10^{-4}$	$7.2\cdot10^{11}$	2.8
Untempered glass	$3\cdot10^{-3}$	$6.7\cdot10^{11}$	2.5

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$$f_0=\frac{1}{2\pi}\sqrt{\frac{E_0S}{M_0h}};\qquad(5.2)$$

-- for designs shown in Fig. 32, c

$$f_0=\frac{1}{2\pi}\sqrt{\frac{G_0S}{M_0h}}.\qquad(5.2a)$$

Here E and G are Young's modulus and shear modulus of the rubber layer of the antivibrator; h is the thickness of this layer; S is the area of contact between mass M and the rubber layer. For designs shown in Fig. 32, a,b,c, this area is respectively:

$$S=\frac{1}{4}\pi D^2;$$

$$S=\frac{1}{4}\pi\left(D^2-D_0^2\right);$$

$$S=\pi Dh.\qquad(5.3)$$

Formulas (5.2) hold true under the condition that $h<\lambda_{0c}/6$ (λ_{0c} is the length of the shear wave in the material of the rubber layer).

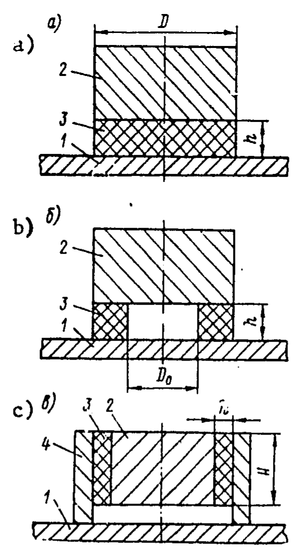

Fig. 32. Structure of an audio frequency antivibrator.

Key: 1. damped plate; 2. metal mass; 3. rubber layer; 4. ring

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It can be seen from formula (5.1) that the loss factor of a plate with an antivibrator η depends on η_U and μ_f. Figure 33 shows the dependence of this factor on μf at various η_U. The maximum value of η decreases with increase in η_U, wherein the frequency band where values η are substantial, expands. Thus when it is necessary to damp a plate with an antivibrator in a wide frequency band a material which has a high loss factor should be chosen for the rubber layer. In order to compensate for the decrease in maximum value of η, the ratio of the mass of the antivibrator M_U to the mass of the plate being damped $M_{ПЛ}$ should be increased.

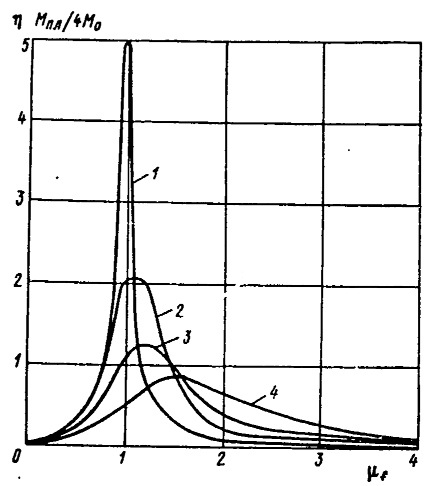

Fig. 33. Dependence of the loss factor of a plate with antivibrator on the ratio $\mu f = f/f_U$ at various η_U.

Key: 1. $\eta_U=0.2$; 2. $\eta_U=0.5$; 3. $\eta_U=1$; 4. η_U-2.

Maximum values $\eta=\eta_{max}$ take place at the frequency

$$f_{max} = f_0 \sqrt[4]{1+\eta_0^2}. \tag{5.4}$$

With rise in η_0 value fmax, compared to f_0 increases. At frequency fmax the loss factor of the plate, when f=fmax is substituted into formula (5.1), takes the form

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$$\eta_{max} = \frac{2M_0\eta_0}{M_{пл}\left(\sqrt{1+\eta_0^2}-1\right)}. \qquad (5.5)$$

The dependence of ηmax on η_U is shown in Fig. 34. It can be seen that ηmax achieves greatest value at $\eta_U<0.2$. However, the damping effect will be manifested only within a narrow frequency band, which for $\eta>\eta max/2$ is [34]:

$$\Delta f \approx f_0\sqrt{2\eta_0\sqrt{1+\eta_0^2}}. \qquad (5.6)$$

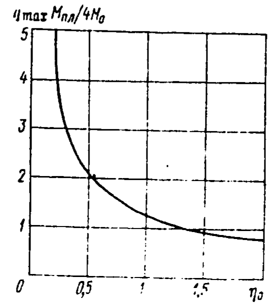

Fig. 34. Dependence of maximum value of loss factor of a plate with antivibrator on loss factor of the rubber layer.

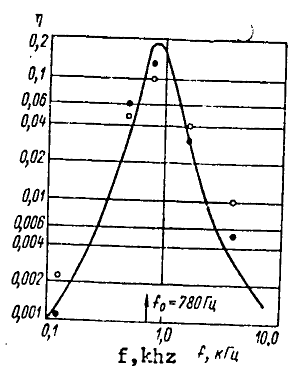

Fig. 35. Frequency relationship of loss factor of a plate with antivibrator.

Key: —— calculation by approximate formula (5.1);
● calculation by precise formula [24];
O experiment

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Figure 35 shows results of correlation of calculated and measured values of the loss factor of a steel plate, 60 cm in diameter and 0.6 cm thick ($M_{пл}$=13 kg), in the center of which an antivibrator was installed (see Fig. 32,a) with characteristics: M_0=368 g, f_0=0.78 khz, η_0=0.6. An approximate calculation was done by formula (5.1). Results of precise calculation and experiment are taken from data in work [24]. Good correlation can be seen between results of the approximate and precise calculation as well as the experiment.

If a second antivibrator were installed on the plate, then the loss factor within it could be determined by the formula

$$\eta=\sum_{i=1}^{n}\eta_i\mu_{\xi i}, \tag{5.7}$$

where η_i is calculated by formula (5.1); $\mu_{\xi i}$ is the ratio of the amplitude of lateral displacement of the plate at the point at which the antivibrator i is installed to the amplitude of the same displacement in the antinode of oscillations of the plate.

By using local vibration absorbers of the type under discussion, high loss factor values can be derived in a limited frequency band with a relatively low ratio of mass of the absorber to mass of the damped plate. For instance, in the calculation just mentioned, where η_{max}=0.19, this ratio amounts to a total of 3%, while in the use of vibroabsorptive coating the same ratio is an order of magnitude greater.

The use of local absorbers to damp vibrations of ship structure plates is recommended primarily at discrete frequencies. They are also used to combat increases in vibration which occur in the process of operation of the structures.

The designs of antivibrators (see Fig. 32, a,b) allow greater values to be achieved at low frequencies. The design shown in Fig. 32,c is preferable from the point of view of operational reliability.

Installation of vibration absorbers on unidimensional rod structures (beams, pipes, etc.), in which there are traveling elastic waves, can give a substantial effect -- up to complete cessation of transfer of vibratory energy [15, 65]. The oscillatory process in a rod takes place because the elastic forces, which occur during deformation of the rod, are counteracted by the inertial resistance of its mass. If elastic resistance, evenly spread along its length, is applied to the rod, which exceeds in its modulus the inertial resistance, then inertial resistance to the elastic forces disappears; the vibratory processes in the rod ceases and, consequently, so does the transfer of vibratory energy. We shall examine the differential

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equation which describes the elastic oscillations of a rod. In the case of longitudinal oscillations this equatiin for the harmonic process may be written as

$$D\zeta'' - (j\omega m + z_P)\, j\omega\zeta = 0, \tag{5.8}$$

where D is the tensile rigidity; ζ is the longitudinal displacement of a section through the rod; m is the mass per unit of length of the rod; z_p is resistances of external load per unit of length of the rod relative to longitudinal forces. Using the antivibrators described above as zp we have

$$z_P = \mathrm{Re}\, z_P + j\, \mathrm{Im}\, z_P, \tag{5.9}$$

where, according to [19],

$$\mathrm{Re}\, z_P = \frac{nM_0\mu_f^3\eta_0}{2\pi f_0\left[\eta_0^2 + (\mu_f^2 - 1)^2\right]};$$

$$\mathrm{Im}\, z_P = \frac{nM_0\mu_f\left[\eta_0^2 + (1 - \mu_f^2)\right]}{2\pi f_0\left[\eta_0^2 + (\mu_f^2 - 1)^2\right]},$$

n is the number of antivibrators per unit of length of the rod. The distance between axes of the antivibrators is substantially less than length of the wave in the rod.

From equation (5.8) it can be seen that if Im z_p has an elastic character and $|\mathrm{Im} z_p| > \omega m$, then this equation where $\eta_0 = 0$ takes the form $\zeta'' - k^2\zeta = 0$. Its solution $\zeta(x) = \zeta_0 \exp(-kx)$ has an attenuating character and attests to the absence of energy transfer along the rod.

Figure 36 shows the dependences of values of the imaginary part of the load on the rod jImzp and intertial resistance of the rod $j\omega m$, taken of the opposite sign, on $\mu_f - f/f_0$. It can be seen from the figure that at a sufficiently low value η_0 the modulus of elastic resistance of the antivibrator Imzp exceeds the modulus of inertial resistance of the rod ωm in some frequency band Δf, situated near the resonant frequency of the antivibrator f_0. Beyond the boundaries of Δf the oscillatory process in the rod has the character of a traveling wave with attenuating amplitude along coordinate x.

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Equation (5.8) can be rewritten as

$$\zeta'' + k^2 (1 - j\eta)\zeta = 0, \qquad (5.10)$$

where $$k^2 = k_n^2 \left(1 + \frac{\operatorname{Im} z_P}{\omega m}\right); \quad \eta = \frac{\operatorname{Re} z_P}{\omega m + \operatorname{Im} z_P};$$

$k_n = \sqrt{\frac{\omega^2 m}{D}}$ is the wave number of longitudinal oscillations of the rod at $z_p=0$.

From equation (5.10) it follows that attenuation of the amplitude of the traveling wave in the rod beyond the boundaries of Δf will be discribed the exponent $\exp(-kx\eta/2)$. This attenuation will decrease as it moves away from the boundary frequencies of Δf. When $\operatorname{Im} z_p < \omega m$ the loss factor of the rod η coincides with the value of the loss factor of a plate with antivibrator, described by expression (5.1), taking into account that $M_{\text{ПЛ}}/4$ is the equivalent mass of the basic oscillations of the plate [44].

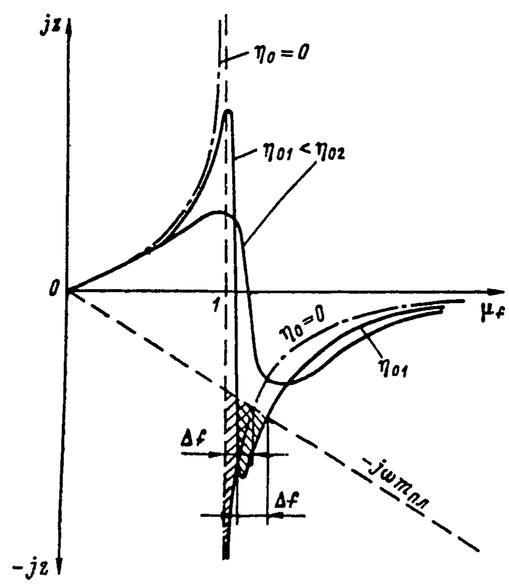

Fig. 36. Dependence of inertial resistance of a rod (---) and elastic component of load resistance Im zp (-.-.-) on $\mu_f=f/f_0$ at various η_0.

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In the case of flexural oscillations of the rod, the differential equation will have the form

$$\xi^{IV} - k^4 (1 - j\eta)\,\xi = 0, \qquad (5.11)$$

where $k^4 = k_{\text{и}}^4 \left(1 + \frac{\operatorname{Im} z_F}{\omega m}\right)$; $\eta = \frac{\operatorname{Re} z_F}{\omega m + \operatorname{Im} z_F}$;

$k_{\text{и}} = \sqrt[4]{\frac{\omega^2 m}{B}}$ is wave number of flexural oscillations of the rod at $z_F = 0$;

z_F is resistance of the load per unit of surface of the rod relative to lateral force F, described by formula (5.9) if index P is replaced by F. It is not difficult to show that all conclusions drawn above for longitudinal oscillations of the rod hold true for it flexural oscillations.

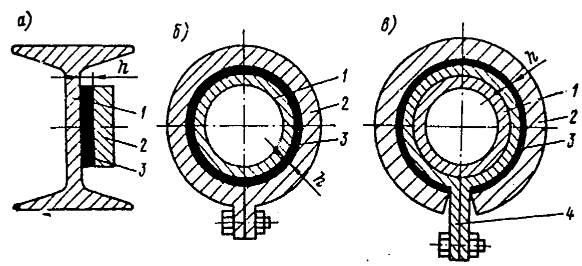

Fig 37. Designs of local vibration absorpbers for rod structures.

Key: 1. damped rod; 2. mass of the vibration absorber; 3. rubber part of the vibration absorber; 4. mounting clamp.

Possible designs for local vibration absorbers for rod structures (including pipes) are shown in Fig. 37. Various types of elastic oscillations of a damped rod can occur at one and the same frequency. Therefore, a design for a local vibration absorber for a rod should be chosen so that frequencies f_0 for longitudianl and torsional oscillations of the rod are always equal to each other.

Values f_0 can be determined by formulas:

-- for lateral displacements of the surface of the rod

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{E_0 S}{M_0 h}}; \qquad (5.12)$$

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-- for tangential displacements of the surface of the rod

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{G_0 S}{M_0 h}}, \qquad (5.13)$$

where E_0 and G_0 are Young's modulus and shear modulus of the material in the rubber part of the vibration absorber; M_0 is the mass of the vibration absorber; S is the area of contact of the vibration absorber with the damped rod; h is the thickness of the rubber part. More detailed information on antivibrators can be found in work [3].

§ 18. Friable Vibroabsorptive Materials

Some friable materials (sand, for example) have an appreciable vibration absorbtion effect [34]. Such materials cannot hold shape; they must, therefore, be used as a filling for hollow structures (for example, tubes, frames, pillars, etc.).

In spite of the absence of cohesive forces between separate particles, friable materials constitute a medium in which elastic waves can propagate. According to data in work [65], in dry sand $c_{\Pi} \sim 1.5 \cdot 10^4$ cm/c, $c_c \sim 10^4$ cm/c. The energy of elastic waves which propagate in sand is absorbed due to friction between its particles. The loss factor which characterizes this absorption is approximately $\eta \sim 0.1$.

Taking the aforesaid into account, it can be concluded that a layer of friable material contiguous to the surface of a vibrating structure behaves much like a pliable vibroabsorptive layer, the properties of which were described in §12. In accordance with these properties, a layer of friable material will begin to effectively absorb vibratory energy from the frequency at which a quarter of the elastic wave excited in the material falls along the thickness of the layer. This frequency f_{p1} is determined by formula (3.48). At frequencies higher than f_{p1} the loss factor of a flexurally-oscillating plate or rod, contiguous to which the material lies, can be calculated approximately by formula (3.52).

Figure 38 shows the frequency characteristics of the loss factor of a steel tube with an outside diameter of 1" and a length of 1.5 mm, filled with sand. The loss factor has appreciable values at frequencies higher than 1.3 khz. By formula (3.48) frequency f_{p1}=1.9 khz. This result agrees satisfactorily with the experiment. Thickness h_{Π}, included in formula (3.48) was taken to be equal to the inside diameter of the tube. Evaluation of the loss factor at frequencies higher than 1.9 khz by formula (3.52) produced values that correspond with experimental values. In the calculation it was assumed that $\rho_{\Pi} \sim 2.2$ and $c_{\Pi} \sim 1.5 \cdot 10^4$cm/c.

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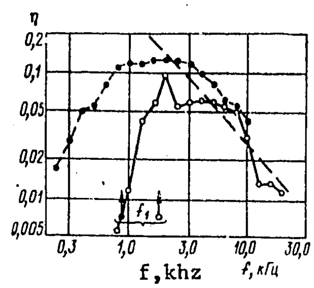

Fig. 38. Loss factor in steel tubes filled with sand.

Key: O– 1" diameter tube
●– 2" diameter tube
-- calculation by formula (3.52)

Table 5

Resonant frequencies of a tube 2" in diameter and 4 m in length

Frequencies of the tube, hz	Number of symmetrical modes of flexural oscillations		
	5	7	9
Hollow (experiment)	101	247	455
Sand-filled (experiment)	84	204	366
Sand-filled (calculation)	79	193	355

According to formula (3.48) an increase in the diameter of the tube must cause a reduction in frequency f_{p1}. The loss factor of a steel tube 2" in diameter filled with sand has values on the order of 0.05–0.1 at frequencies from 0.4 khz (Fig. 38).

In following a hollow structure with friable materials a question arises as to the change in their resonant properties due to increase in mass. Figure 39 shows results of measurements of the mechanical conductivity of a steel tube 2" in diameter and 4 m in length both with and without sand within it (measurements made by V.C. Konevalov and V.V. Moiseyev). The tube was suspended by its ends on elastic fasteners and excited by lateral force in the center. After the

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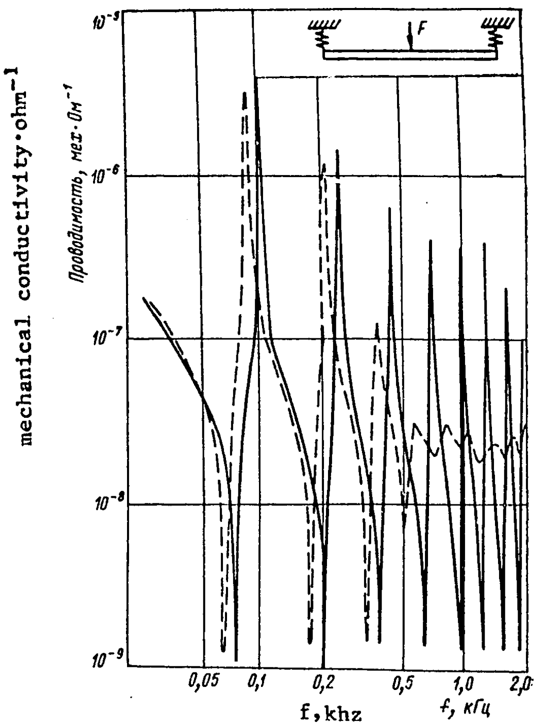

Fig. 39. Mechanical resitance of a steel tube 2" in diameter and 4 m in length.

Key: ——tube without sand
---tube filled with sand

tube was filled with sand its resonant frequencies decreased. Table 5 shows values of resonant frequencies of a few symmetrical modes of flexural oscillations of the tube being studied, derived by experiment, proceeding from an increase in mass of the tube and no change in its flexural rigidity when filled with sand.

It can be seen that the expected decrease in resonant frequencies corresponds well with results of the experiment. Thus the value of the resonant frequency $f'_{рез}$ of a hollow structure filled with a friable material can be calculated by the formula

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$$f'_{\text{рез}} = f_{\text{рез}} \sqrt{\frac{m}{m'}}, \qquad (5.14)$$

where fpe3 is the resonant frequency of the structure before it is filled; m, m is mass the structure per unit of length before and after filling respectively.

Introducation of vibroabsorptive material into a structure substantially moderates the characteristics of its vibroexcitability. Rammed fine (diameter up to 0.5 mm) cast iron shot, which is used for blast-cleaning in shipbuilding yards, can be used for the friable vibroabsorptive material. The vibroabsorptive effect of this material is almost the same as that of sand. However, cast iron shot is less dangerous from the point of view of abrasive action should it accidentally get into contacting machine parts.

On flat horizontal structures friable materials are convenient for experimental determination of the feasibility of damping finished structures. As studies showed, a double-thickness layer of any friable vibroabsorptive material is sufficient to achieve a significant loss factor (studies by I.I. Klyukin and A.I. Kurbatov).

§ 19. Liquid Intermediate Layers Used for Vibration Absorption

A structure consisting of two plates, with the space between them filled with a viscous liquid (Fig. 40), has vibroabsorptive properties [82]. With excitation in one of the plates of a flexural wave two modes (forms) os oscillations can occur in the structure: symmetrical and antisymmetrical [81]. Deformations of the structure, corresponding to these modes, are shown in Fig. 40. In the symmetrical mode oscillations of the plate occur in antiphase; in the antisymmetrical mode thse oscillation are cophasal.

In work [81] it is shown that the symmetrical mode is characterized by greater loss factor values and a lower velocity of propagation of oscillations. Therefore, oscillations in the structure being examined, which correspond to this mode, attenuate very rapidly. Consequently, only the antisymmetrical mode has a practical value.

Due to the relationship existing between the plates, the wave number of flexural oscillations in them is identical. If it is supposed that the amplitude of oscillations of both plates is on the same order and the mass of the liquid is disregarded, then the indicated wave number will be

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$$k_{\text{н}} \approx \sqrt{\omega}\,\sqrt[4]{\frac{m_1+m_3}{B_1+B_3}}. \qquad (5.15)$$

Here and further on indices 1, 3 and 2 apply to values which characterize the plates and the layer of viscous liquid respectively. From expression (5.15) we derive the value of the phase velocity of flexural waves in the structure being examined:

$$c_{\text{н}}^2=\frac{\omega h_1 c_{\text{п}1}}{\sqrt{12}}\sqrt{\frac{1+\alpha_{31}^3\beta_{31}^3}{1+\alpha_{31}\gamma_{31}}}=c^2_{\text{н}1}\sigma, \qquad (5.16)$$

where

$$\alpha_{31}=\frac{h_3}{h_1};\quad \beta_{31}=\frac{E_3}{E_1};\quad \gamma_{31}=\frac{\rho_3}{\rho_1};\quad c_{\text{н}1}=\sqrt{\frac{\omega h_1 c_{\text{п}1}}{\sqrt{12}}}.$$

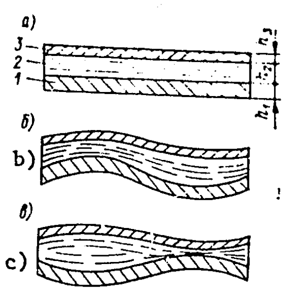

Fig. 40. Vibration absorbing structure with liquid intermediate layer (a) and the character of its deformation in antisymmetrical (b) and symmetrical (c) modes.

Key: 1. damped plate; 2. viscous liquid; 3. attached plate.

With plates of the same material, which is often the case in practice, $\beta_{31}=\gamma_{31}$ 1 and $\sigma=(1+\alpha^3_{31})^{1}2$ $(1+\alpha_{31})$; coefficient σ has maximum deviation from one: $\sigma=0.865$ at $\alpha_{31}=0.5$ ($0<\alpha_{31}$ 1). Therefore, for practical evaluations it can be assumed that $c_H \sim c_{H1}$, wherein the greatest error will be less than 7%.

We shall determine the loss factor of a flexurally-oscillating plate with thickness h_1, to which a plate with thickness h_3 is attached with a spacing h_2, which is filled with a viscous liquid with a dynamic viscosity factor of μ_2 and density ρ_2. In doing so we shall assume that the layer of liquid transmits only lateral forces from one plate to the other, We shall determine the loss factor by the wave mechanical resistance method (see $9), according to which

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$$\eta = \frac{\operatorname{Re} z_{23}}{\omega m_1}, \tag{5.17}$$

where $\operatorname{Re} z_{23}$ is the real part of the acoustical load on plate 1 relative to lateral forces which, according to work [81], is

$$\operatorname{Re} z_{23} = \frac{a_2 (a_3 - b_3)^2}{a_2^2 + (a_3 - b_3 + b_2)^2}, \tag{5.18}$$

where

$$a_2 = \frac{12 \mu_2 c_н^2}{\omega^2 h_2^3};$$

$$a_3 = \omega m_3;$$

$$b_2 = \frac{\rho_2 c_н^2}{\omega h_2};$$

$$b_3 = \frac{B_3 \omega^3}{c_н^4}.$$

Let us introduce the notations

$$\gamma_2 = \frac{12 \mu_2}{h_2 m_2}; \quad \beta_{21} = \frac{h_1 c_{п1} m_2}{\sqrt{12}\, h_2^2}. \tag{5.19}$$

Taking these notations into account, from expression (5.17), after simple conversion, we have

$$\eta = \frac{1}{\frac{m_1}{m_3^2 \alpha^2} \left[\frac{\gamma_2 \beta_{21}}{\omega} + \frac{1}{\gamma_2 \beta_{21}} (\omega m_3 \alpha + \beta_{21})^2 \right]}, \tag{5.20}$$

where $\alpha = 1 - \alpha_{31}^2 \beta_{31}^2 \gamma_{31}^2;\ (\alpha = \alpha_0 = 1 - \alpha_{31}^2 \text{ при } \beta_{31} = \gamma_{31} = 1).$

The result obtained differs from data presented in work [81], since the work examined only the case where $h_3 < h_1$.

At low frequencies ($\omega \to 0$)

$$\eta \to \frac{\omega^2 m_3^2 \alpha^2}{m_1 \gamma_2 \beta_{21}} = \frac{\rho_3^2 h_3^2 h_2^3 \alpha^2 \omega^2}{\sqrt{12}\, \rho_1 h_1^2 c_{п1} \mu_2}. \tag{5.21}$$

At high frequencies ($\omega \to \infty$)

$$\eta \to \frac{\gamma_2 \beta_{21}}{\omega^2 m_1} = \frac{\sqrt{12}\, c_{п1} \mu_2}{\omega^2 \rho_1 h_2^3}. \tag{5.22}$$

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From expressions (5.21) and (5.22) it can be seen that with increase in frequency the loss factor of the structure first rises and then drops. Consequently, at some frequency ω_{max} the loss factor reaches maximum. Frequency ω_{max} can be found from the equation

$$\omega_{max}^3 m_3\alpha(\omega_{max}m_3\alpha+\beta_{21})-\gamma_2^2\beta_{21}^2=0. \qquad (5.23)$$

Approximate solution to this equation appears as

$$\omega_{max}\approx\sqrt{\frac{\gamma_2\beta_{21}}{m_3\alpha}} \quad (\omega_{max}<\gamma_2); \qquad (5.24)$$

$$\omega_{max}\approx\sqrt[3]{\frac{\gamma_2^2\beta_{21}}{m_3\alpha}} \quad (\omega_{max}>\gamma_2). \qquad (5.25)$$

It will be noted that error in calculation of ω_{max} by formulas (5.24) and (5.25) does not exceed 19% (at $\omega_{max}=\gamma_2$). Substituting ω_{max} values into formula (5.20) we find that

$$\eta_{max}\approx\frac{1}{\frac{m_1}{m_3\alpha}\left(2+\frac{2\omega_{max}}{\gamma_2}+\frac{\omega_{max}^2}{\gamma_2^2}\right)} \quad (\omega_{max}<\gamma_2); \qquad (5.26)$$

$$\eta_{max}\approx\frac{1}{\frac{m_1}{m_3\alpha}\left(\frac{3\omega_{max}}{\gamma_2}+\frac{\gamma_2}{\omega_{max}}+\frac{\omega_{max}^3}{\gamma_2^3}\right)} \quad (\omega_{max}>\gamma_2). \qquad (5.27)$$

Where $\omega_{max}=\gamma_2$ both expression for η_{max} agree. Figure 41 shows the dependence of η_{max} on the ratio γ_2/ω_{max}. With increase in γ_2/ω_{max} values of η_{max} rise. It is not advantageous to design a structure with $\gamma_2/\omega_{max}>3$, since this does not give a substantial gain in the quantity η_{max}.

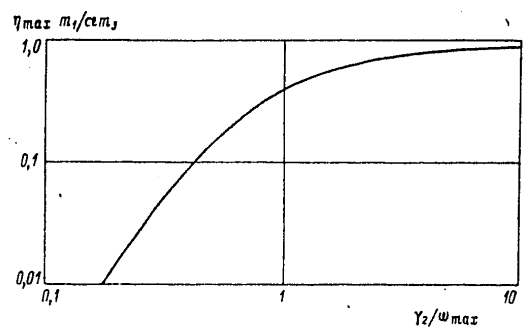

Fig. 41. Dependence of maximum loss factor of a structure with liquid intermediate layer on the ratio γ_2/ω_{max}.

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Let us analyze the dependence of the loss factor in a structure η on the ratio of thicknesses of the plates which compose it $\alpha_{31}=h_3/h_1$. From formulas (5.21) and (5.22) it follows that this dependence occurs at frequencies near ω_{max} and at lower frequencies. At high frequencies η does not depend on α_{31}.

The indicated dependence at low frequencies, as follows from formula (5.21), is described by parameter α, equal to ($\beta_{31}=\gamma_{31}=1$)

$$\alpha=\alpha_0=\alpha_{31}\left(1-\alpha_{31}^2\right)^2. \tag{5.28}$$

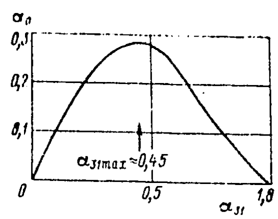

Fig. 42. Dependence of α_0 on α_{31}.

Figure 42 shows the dependence of α_0 on α_{31}. The greatest values of α_0, and consequently also the loss factor of the structure, will be reached at $\alpha_{31max}=0.45$.

With increase in α_{31}, simultaneous with rise at low frequencies of the loss factor of the structure, ω_{max} will decrease. This decrease will occur with an increase of α_{31} to value 0.45. At greater values of α_{31}, rise in ω_{max} begins (Fig. 43,a).

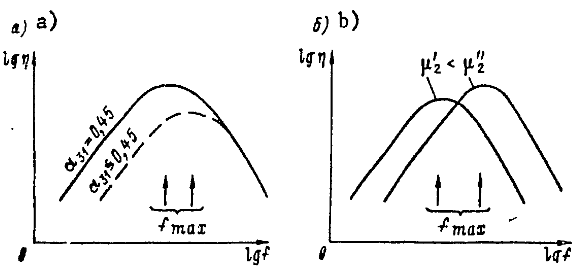

Fig. 43. Frequency characteristics of the loss factor of a structure with liquid intermediate layer at various values of α_{31} (a) and μ_2 (b).

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Losses of vibratory energy in the structure under examination are attributable to movement of the viscous liquid in the space between plates anlong their plane. This movement occurs as a result of changes in the spacing under flexural oscillations of the plates (see Fig. 40). The amplitude of movement of the liquid, and consequently also losses of vibratory energy, will be greater as the amolitudes of flexural oscillations of the basic plate 1 and the attached plate 3 differ more significantly. From this point of view the shape of the curve in Fig. 43,a can be explained. With identical plates (α_{31}=1) the wave mechanical resistance of the attached plate z_3=ja_3-jb_3 is equal to zero (a_3=b_3), and flexural oscillations are easily excited within it. The amplitudes of oscillations of both plates are in this case identical and η=0. With decrease in thickness of plate 3 (α_{31}<1) the wave resistance of this plate at first increases due to mismatch of its inertial and elastic components (a_3>b_3). In this case the amplitude of oscillations of plate 3 decrease in comparison with the amplitude of the oscillations of plate 1 and η increases. With further decrease of α_{31}, the inertial part of wave resistance in plate 3 will begin to predominate a_3=ωm_3(a_3>b_3). Therefore wave resistance drops with decrease of h_3, and consequently also m_3.

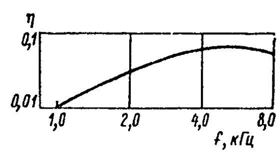

Fig. 44. Loss factor of a structure of steel plates 3 and 2 mm thick with an intermediate layer of synthetic liquid rubber SKN-26, calculated by formula (5.20).

Accordingly the difference in amplitudes of oscillations of the plates making up the structure decreases and, consequently, so does η.

From formulas (5.24) and (5.25) it can be seen that an increase in the viscosity of the liquid μ_2 in the space between the plates raises the frequency fmax, at which the loss factor of the structure has maximum value (Fig. 43,b). By selection of values of the viscosity of the liquid μ_2 the required value fmax can be achieved. Liquids with a dynamic viscosity of 10^2-10^4 DIN·c/m^2 are suitable for use in the subject structure.

Their basic characteristics are shown in Table 6.

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Table 6

Physio-Mechanical Characteristics of Liquid at Temperature About 20°C

Liquid	ρ, g/cm^3	μ, DIN/cm^2
Water	1.0	10-2
Glycerine	1.26	8.5
Castor oil	0.96	10.3
Silicone oil	1.0	80.0
Synthetic rubber SKN-26	1.6	$5.3 \cdot 10^3$

It should be kept in mind that the results obtained are suitable only for moderate values of viscosity, at which shear forces are not transmitted from one plate to the other. At greater values of viscosity calculations of the loss factor of a structure with a liquid intermediate layer by the formulas given in this paragraph will give values that are too low. From Fig. 44 it is evident that effective damping can be achieved ($\eta \sim 0.1$) by the use of liquid intermediate layers.

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Chapter 6. DAMPING OF VIBRATIONS OF ELEMENTS OF SHIP MACHINERY AND HULL-FRAME STRUCTURES

§ 20. Optimum Length of Vibroabsorptive Coatings

Under actual ship conditions, where one of the basic problems is the limitation on displacement, it is important to find the optimum variant for placement of vibroabsorptive coatings, which will yield maximum acoustical effect for a given weight. Such an optimization is possible because as the length of plates in hull-frame structures, that are to be faced with coatings, varies so does the thickness of coatings and, consequently, the loss factors.

We shall determine the optimum length for various types of vibroabsorptive coatings on characteristic ship hull-frame structures, namely, on a uniform plate and a plate reinforced with periodic rigidity ribs. We shall assume that either unidimensional (flat) flexural waves or bidimensional (cylindrical) waves propagate in the plates. As this takes place there is a sector of the plate with length $L_{BП}$, faced with a vibroabsorptive coating, which lies in the path of propagation of the flat wave. And around the source, which excites the cylindrical wave, there is also a coating applied to a sector of plate with radius $L_{BП}$. The first case corresponds to application of a coating a distance away from the source of vibration (machinery); the second to application in direct proximity to the source. Patterns for applying coatings, corresponding to the indicated variants, are shown in Fig. 45.

It is shown in § 28 that attenuation of amplitude of a flexural wave in a plate does not depend on its spatial characteristics and is determined by the length of the coating. According to formula (7.5), the effectiveness of a vibroabsorptive coating on a uniform plate Э, in db is

$$Э \approx 2,15 k_{\text{изл}} \eta L_{\text{вп}}, \qquad (6.1)$$

where η is the loss factor of a plate faced with the coating. For a ribbed plate the analogous expression has the form

$$Э \approx 4,3 \sqrt{\frac{k_{\text{изл}} \eta}{\alpha_0 l_0}} L_{\text{вп}}, \qquad (6.2)$$

where l_0 is the distance between rigidity ribs reinforcing the plate; α_0 is the coefficient of transmission of energy of flexural waves through the rigidity with diffuse incidence.

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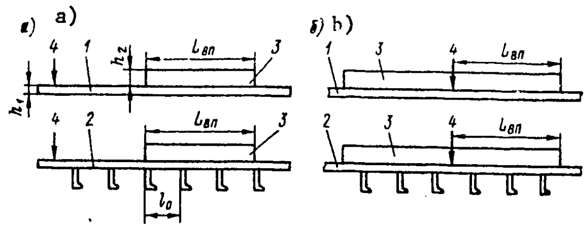

Fig. 45. Patterns for application of a vibroabsorptive coating on a hull-frame structure: a. for flat wave; b. for cylindrical wave.

Key: 1. uniform plate; 2. ribbed plate; 3. coating; 4. source of vibration.

It can be seen from expressions (6.1) and (6.2) that maximum effectiveness will correspond to the greatest value of parameters $\eta L_{B\Pi}$ and $\sqrt{\eta} L_{B\Pi}$. Insofar as η and $L_{B\Pi}$ at a given mass of the coating $M_{B\Pi}$ are functions of the thickness of the coating h, our problem amounts to finding the solution to equations:

-- for a uniform plate

$$\frac{\partial[\eta(h) L_{\text{вп}}(h)]}{\partial h}=0; \tag{6.3}$$

-- for a ribbed plate

$$\frac{\partial[\sqrt{\eta(h)} L_{\text{вп}}(h)]}{\partial h}=0, \tag{6.4}$$

in which parameters $\eta L_{B\Pi}$ and $\sqrt{\eta} L_{B\Pi}$ have maximum values.

We shall begin examination of the problem with a rigid vibroabsorptive coating, for which ratios between coating length $L_{B\Pi}$ and coating thickness h_2 are as follows:

--for a flat wave

$$M_{\text{вп}}=L_{\text{вп}} H_{\text{вп}} \rho_2 h_2; \tag{6.5}$$

-- for a cylindrical wave

$$M_{\text{вп}}=\pi L_{\text{в}}^2\, \rho_2 h_2, \tag{6.6}$$

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where ρ_2 is density of the coating material; $H_{BП}$ is the width of the faced plate. The loss factor of the coating is determined by formula (3.31). By substituting formulas (3.31), (6.1), (6.2), (6.5) and (6.6) into expressions (6.3) and (6.4) it is not difficult to derive an equation for determining $\alpha_{2_{ОПТ}}$, at which the effectiveness of the coating reaches maximum:

where

Values of coefficient n are as follows:

In a uniform plate for waves:
- flat.................3
- cylindrical..........6

In a ribbed plate for waves:
- flat.................3/2
- cylindrical..........3

Dependences of β_2 on $\alpha_{2_{ОПТ}}$ at the indicated values of n are shown in Fig. 46. Figure 47 shows dependences of effectiveness Э on α_2.

Analysis of Figures 46 and 47 show the following.

In the first case (n=3) at $\beta_2 > {}^2/_3$ the dependence of Э on α_2 (h_2) has no maximums, asymptotically approaching final and zero value when $\alpha_2 \to 0$ and $\alpha_2 \to \infty$. At the $\beta_2 < {}^2/_3$ practically used in rigid vibroabsorptive coatings (see $ 10), Э reaches maximum at $\alpha_{2_{ОПТ}}$, defined for a given β_2 (see Fig. 46).

In the second case (n=6) Э tends toward zero at $\alpha_2 \to 0$ and $\alpha_2 \to \infty$ (Fig. 47). Maximum effectiveness exists at any β_2. The value of $\alpha_2 = \alpha_{2_{ОПТ}}$, which corresponds to this maximum is determined from Fig. 46.

In the third case $n = {}^2/_3$) has no solution when $\beta_2 > {}^2/_3 \cdot 10^{-2}$. Accordingly, at the indicated values β_2 effectiveness of the coaing has no extreme values and approaches infinity at $\alpha_2 \to 0$ (Fig. 47). In this case the effectiveness will be more significant as α_2 is less and, therefore, as the coating length is greater. The maximum attainable length of the coating is determined in this case by the dimensions of the structure being damped.

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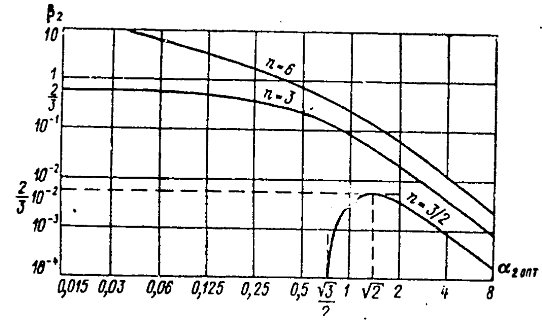

Fig. 46. Dependence of β_2 on $\alpha_{2ОПТ}$ at various .

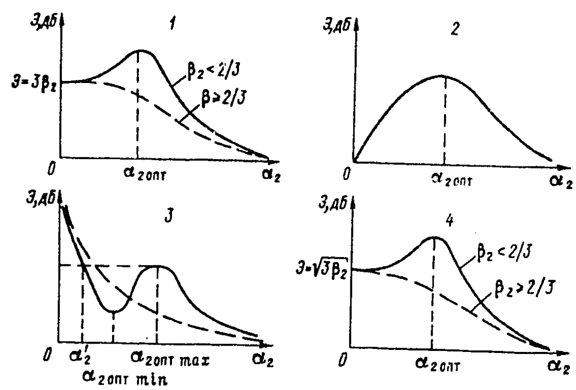

Fig. 47. Dependences of the effectiveness of vibroabsorptive coatings on $\alpha_2=h_2/h_1$.

Key: 1. Flat wave in a uniform plate; 2. cylindrical wave in a uniform plate; 3. flat wave in a ribbed plate; 4. cylindrical wave in a ribbed plate.

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When $\beta 2<{}^{2}/_{3}\cdot 10^{-2}$ the effectiveness of the coating has two extreme values. One of these ($\alpha 2=\alpha 2_{\text{ОПТ}}$ min) corresponds to minimum effectiveness; the other ($\alpha 2=\alpha 2_{\text{ОПТ}}$ max) to maximum.

When choosing $\alpha 2$ one should strive to meet the condition $\alpha 2<\alpha 2'$ (see Fig. 47). If the dimensions of the structure being damped are such that this condition cannot be met, the greatest effect will be realized when $\alpha 2=\alpha 2_{\text{ОПТ max}}$.

In the fourth case (n=3) the aforesaid relative to the first case holds true. The only difference is that at $\alpha 2\to 0$ the effectiveness of the coating tends toward another asymptotic value (see Fig. 47).

A rigid vibroabsorptive coating from the plastic "Agat", which is used in shipbuilding (for steel damped structures $\beta 2=5\cdot 10^{-3}$), according to the results obtained, must have a thickness which exceeds the thickness of the plate being damped by factors of 4, 5.6, 2 and 4 for the cited cases respectively. Figure 48 shows the effectiveness of such a coating on a uniform plate, dependent on the ratio $\gamma=L_{\text{ВП}}/L_{\text{ВП ОПТ}}$ ($L_{\text{ВП ОПТ}}$ correspond to optimum values of $\alpha 2$. The following were used in the calculation: $h1=0.5$ cm, $\rho 1=7.8$ g/cm, $L_{\text{ВП}}=200$ cm, $\eta 2=0.33$, $\rho 2=1.8$ g/cm^3, $f=10^3$ hz. From Fig. 48 it can be seen that deviation from optimum values of $L_{\text{ВП}}$ can substantially decrease the effect of using a vibroabsorptive coating. It should be kept in mind that when $\alpha 2>1$ the effectiveness of the vibroabsorptive coating can be adversely affected by shear deformations which develop within it at frequencies when $k_c 2h2>1$ (see § 13). Therefore, the actual values of $\alpha 2$, at which maximum Э occurs, can lie somewhat lower than values obtained without taking these deformatins into account.

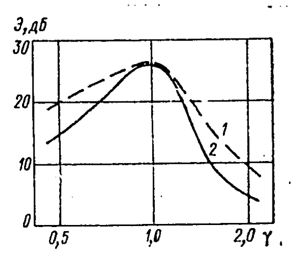

Fig. 48. Dependence of effectiveness of a rigid vibroabsorptive coating of "Agat" plastic, applied to a uniform plate, on $\gamma=L_{\text{ВП}}/L_{\text{ВП ОПТ}}$.

Key: 1. Flat flexural wave; 2. Cylindrical flexural wave

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When determining optimum length for a stiffened vibroabsorptive coating, the extreme character of the frequency dependence of its loss factor must be kept in mind. It is advisable that the chosen frequency $f_{ОПТ}$, at which the loss factor of such a coating reaches maximum (see § 11), be left fixed. According to formula (3.42) the condition for invariability of $f_{ОПТ}$ is the equation

$$\alpha_2\alpha_3 = b_0 = \frac{b}{f_{опт}} = \text{const}, \tag{6.8}$$

where

$$b \approx \frac{G_2 c_{п1}\sqrt{1+\eta_2^2}}{2\pi\sqrt{12}\,E_3 h_1}, \tag{6.9}$$

$f_{ОПТ}$ is determined by formula (3.42). Indices in formula (6.9), corresponding to structures of stiffened coatings, are depicted in Fig. 16,a. For real structures of this coating parameter b has a value on the order of one; therefore, b 1 for practically important values $f_{ОПТ}\sim 1.0$ khz. In determining parameter b it was assumed that the geometric parameter of the coating $\gamma<1$. Taking formula (3.39) into account, with the same material used for the damped plate and the stiffening sheet $\left(\beta_3 = \frac{E_3}{E_1} = 1\right)$

and and allowing for condition (6.8)

$$\gamma \approx 12\left(\frac{1}{2}+\frac{b_0}{\alpha_3}\right)^2 \alpha_3, \ll 1, \tag{6.10}$$

where $\alpha_3=h_3/h_1<1$.

With these allowances, optimum parameters for a stiffened coating can be derived by analogy to rigid coatings. Specifically, for the case of propagation of a flat wave through a uniform plate we have

$$\alpha_{3опт} \approx \frac{\gamma_{23}}{2}; \quad \alpha_{2опт} \approx \frac{2b_0}{\gamma_{23}}, \tag{6.11}$$

where $\gamma_{23}=\rho_2/\rho_3$. The optimum length of a stiffened coating in this case is equal to

$$L_{вп\ опт} = \frac{M_{вп}}{H_{вп} h_1\left(\rho_2 b_0 \alpha_3^{-1} + \rho_3\alpha_3\right)}. \tag{6.12}$$

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Optimum length for a pliable vibroabsorptive coating at frequencies, higher that the frequency of first resonance of this coating (see $ 12), is determined by the dimensions of the structure being damped, since at these frequencies the loss factor of the coating is practically independent of its thickness. The thickness of a pliable coating in this case can be such that frequency f_{p1}, found by formula (3.48), exceeds the lower frequency of the spectrum in which effective damping of the structure is required. In this case the length of the coating should be decreased and its thickness increased in order that the indicated frequencies are equal.

$ 21. Loss Factor in Plates Partially Faced with Vibroabsorptive Coating

Sometimes, for technological or other reasons, a vibroabsorptive coating cannot be applied to the entire surface of a plate to be damped. One the other hand, if the loss factor of a coating depends on its thickness, then how the material should be applied must be specified: either uniformly aver the entire surface of the plate or concentrated on a certain sector of the surface. In both cases it is necessary to know the dependence of the loss factor of a plate that is partially faced with a coating.

Let us suppose that the amplitudes of vibrations of the plate are practically identical over its entire surface, to include when a coating is applied to part of its surface. Then the loss factor of the plate with area S, partially faced with a vibroabsorptive coating, according to formula (3.5) will be

$$\eta \approx \frac{\eta_0 S_0}{S}, \tag{6.13}$$

where η_0 is the loss factor of the coated part of the plate and S_0 is the area of the coated part of the plate.

In the case where a figid vibroabsorptive coating is used the loss factor of the plate depends on the ratio of the thickness of the covering and the plate [see formula (3.31)]. Let us find in connection with this the optimum value S_0, at which loss factor η achieves maximum. Assuming the mass of the covering material to be specified, we have

$$h_2' = \frac{h_2}{\mu_S}, \tag{6.14}$$

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where h_2 and h_2' are thicknesses of the coating in complete and partial facing of the plate (Fig. 49),

$$\mu_S = \frac{S_0}{S}. \qquad (6.15)$$

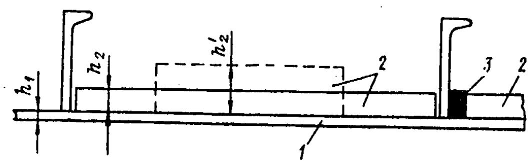

Fig. 49. Schematic of a structure damped with a rigid vibroabsorptive coating.

Key: 1. damped ribbed plate; 2. rigid coating; 3. epoxy filler

Substituting into formula (6,13) the expression for loss factor of a rigid coating with thickness h_2', to be determined by formula (3.31), differentiating the result by μ_s and equating the derived expression to zero, it is not difficult to derive an equation for optimum value μ_s

$$\beta_2 = \frac{3\left(1 + 2\frac{\alpha_2}{\mu_{S\,\text{опт}}}\right)^2}{\frac{\alpha_2}{\mu_{S\,\text{опт}}}\left(3 + 6\frac{\alpha_2}{\mu_{S\,\text{опт}}} + 4\frac{\alpha_2^2}{\mu_{S\,\text{опт}}^2}\right)^2}, \qquad (6.16)$$

where $\alpha_2 = h2/h1$. Fig. 50 shows the dependence of ratio $\alpha_2/\mu_{s\,\text{ОПТ}}$ on $\beta_2 = E_2/E_1$ Specifically, for "Agat" plastic applied to a steel plate ($\beta_2 = 5\cdot 10^{-3}$), $\alpha_2/\mu^{-1}{}_{s\,\text{ОПТ}} = 4$. In this case when $\alpha_2 = 2$ the optimum ratio of the area of the coated part of the plate to the total area is

$$\mu_{S\,\text{опт}} = \frac{S_{\text{опт}}}{S} = 0{,}5. \qquad (6.17)$$

In this case the gain in loss factor in the plate is determined by the formula

$$\mu_\eta = \frac{\eta'}{\eta} = \frac{\mu_{S\,\text{опт}}\nu'(1+\beta_2\nu)}{\nu(1+\beta_2\nu')}, \qquad (6.18)$$

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where $$\nu=\alpha_2(3+6\alpha_2+4\alpha_2^2);$$

$$\nu'=\frac{\alpha_2}{\mu_{S\,\text{опт}}}\left(3+6\frac{\alpha_2}{\mu_{S\,\text{опт}}}+4\frac{\alpha_2^2}{\mu_{S\,\text{опт}}^2}\right),$$

amounts to a factor of 5.6.

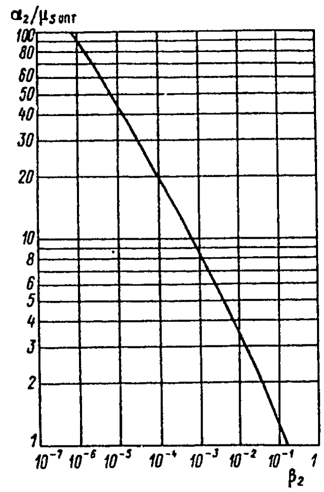

Fig. 50. Dependence of the optimum area to be covered when a plate is damped by a rigid vibroabsorptive coating on $\beta_2=E_2/E_1$.

In practice such a significant increase in effectiveness of a rigid coating will not occur due to the approximation of formula (6.13) and the so-called edge effect. This effect, the results of studies of which are presented in work [45], manifests itself in the fact that at the edges of a coating, uder flexure of a damped plate, instead of strech deformation mainly shear deformation takes place, under which significantly less energy is absorbed in rigid materials. Obviously, the influence of the edge effect is proportional to the length of the free edges of the coating. In view of this, when a plate is to be partially faced with a rigid coating it is more rational to place it in the central part of the plate and not along its edges. Influence of the edge effect can be decreased by smearing an epoxy spackle-filler on the edges of the coating which will be rigid after drying (see Fig. 49). In order to preclude an increase in the influence of the edge effect, it is recommended that rigid coatings not be applied in separate pieces. If dimensions of separate pieces of this coating are less than half the length of the flexural wave in the damped plate, then its effectiveness will be reduced by more than 10% [48].

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For a stiffened vibroabsorptive coating the optimum value μ_s can be derived by fixing an invariable $f_{ОПТ}$, at which the loss factor of the coating η is maximum. By analogy to the way this was done in § 20, we get

$$\mu_{S\,\text{опт}} \approx \frac{2\alpha_3\rho_3}{\rho_2}, \qquad (6.19)$$

where $\alpha_3 = h_3/h_1$ is the ratio of the thickness of the stiffening layer and the damped plate when the coating material is spread over the entire surface of the plate.

If $\mu_{SОПТ}$, calculated by formula (6.19), should be greater than one, then $\mu_{SОПТ}=1$ should be used.

In the use of a stiffened coating its dimensions are of great importance. Work [97] shows that the optimum dimensions of a coating $L_{ОПТ}$ is

$$L_{\text{опт}} = 3{,}28\sqrt{\frac{h_2h_3E_3}{G_2}}. \qquad (6.20)$$

At this value $L_{ОПТ}$ the edge effect, which plays a positive role for a stiffened coating, where absorption of energy is attributable to shear deformation of the viscoelastic material, becomes more intense.

Figure 51 shows the dependence of the loss factor of a stiffened coating on the ratio $L/L_{ОПТ}$. If L is three times greater than $L_{ОПТ}$, then the loss factor of a plate with a stiffened coating decreases by the same factor.

Value $L_{ОПТ}$ corresponds approximately to $\lambda_И/4$ of the damped plate at frequency $f_{ОПТ}$.

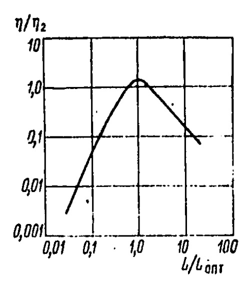

Fig. 51. Dependence of the loss factor of a plate with a stiffened vibroabsorptive coating on the ratio $L/L_{ОПТ}$[97].

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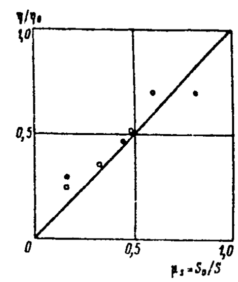

Fig. 52. Dependence of the loss factor of a plate with pliable vibroabsorptive coating on the relative area coated μs.

Key: ___ calculation by formula (6.13)
O coating applied alon edges of plate
● coating applied in middle of plate

When a pliable vibroabsorptive coating is used at frequencies where it is most effective, where the loss factor is practically independent of thickness of the coating, partial facing of a plate to be damped will serve only to decrease its loss factor. Figure 52 shows dependence of the loss factor of a plate on s=S /S, derived by calculation by formula (6.13) and those derived experimentally. The direct proportionality of the loss factor of a damped plate to the relative area that is faced with a vibroabsorptive coating, used as the basis for formula (6.13), is satisfactorily confirmed by experiment.

$ 22. Vibration Absorption in Ribbed Structures

Most ship structures consist of plates reinforced by rigidity ribs. Therefore, determination of the effectiveness of a vibroabsorptive coating applied to such a plate must take into account the influence of the rigidity ribs on the oscillatory properties of the structure. To clarify this influence, let us examine a plate on which a system of parallel equidistant ribs is placed, forming spaces with width L (Fig. 53).

According to formula (7.5) the effectiveness of a vibroabsorptive coating applied to a plate is proportional to the derivative $k_H\eta$ (k_H is the wave number of flexural oscillations in the plate; η is the loss factor, introduced to it by the coating). Wherein

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$$k_{\text{и}} = \sqrt[4]{\frac{\omega^2 m}{EI}}. \tag{6.21}$$

We shall determine how the rigidity ribs influence values k and .

At low frequencies ($f<f_{p1}$, f_{p1} is the first resonant frequency of flexural oscillations of the space) a ribbed plate becomes orthotropal. With flexure of such a plate, in a plane parellel to the rigidity ribs, its oscillatory properties are equal to properties of a rod with width L, cut from the plate along the ribs. A cross section of this rod, shown in Fig. 53 by the solid line, shows an asymmetrical "I" profile.

At frequencies $f<f_{p1}$ the rigidity ribs increase the mass of the plate and its flexural rigidity. Accordingly, the mass per unit of length of the rod being examined will be equal to

$$m_{\text{ст}} = m_L + m_{\text{р}} = m_L(1+\mu_m), \tag{6.22}$$

where $m_L=m_{\text{ПЛ}}L$; $\mu_m=m_p/m_L$; $m_{\text{ПЛ}}$ is the mass of the plate without rigidity ribs which falls within a unit of surface; m_p is the mass of the rigidity rib per unit of length.

Flexural rigidity of the rod is determined by the position of the neutral plane, which will be displaced relative to the plate by distance x. (see Fig. 53).

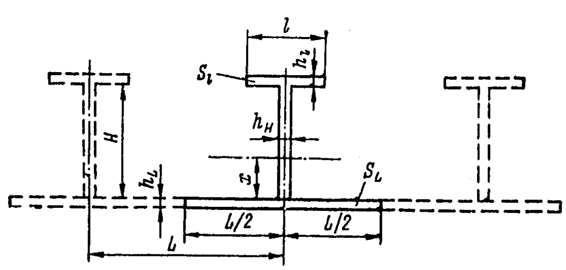

Fig. 53. Ribbed plate.

Value x can be approximately determined, assuming that the flexural rigidity of the I section is betermined by the inertial moment of its shelves. Proceeding from this assumption it is not difficult to get

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$$x \approx \frac{H}{1+\sqrt{\mu_S}}, \qquad (6.23)$$

Where H is the height of the rigidity rib; $\mu_S = \frac{S_L}{S_l}$; S_l, S_L — are area of section through shelves of the I-beam (Fig. 53).

It can be seen from formula (6.23) that the neutral plane will pass close to the base of the rigidity rib, since $S_L > S_1$. With formulas (6.21) and (6.23) we find for low frequencies

$$k_{\text{и нч}} \approx \sqrt[4]{\frac{\omega^2 m_{\text{пл}}(1+\mu_m)(1+\sqrt{\mu_S})^2}{2EH^2 h_L}}. \qquad (6.24)$$

Taking expressions (6.21) and (6.24) into account, the ratio $k_{\text{И НЧ}}/k_{\text{И ПЛ}}$, with which we are concerned, is equal to:

$$\frac{k_{\text{и нч}}}{k_{\text{и пл}}} = \sqrt[4]{\frac{h_L^2(1+\mu_m)(1+\sqrt{\mu_S})^2}{24H^2}}. \qquad (6.25)$$

Let us calculate the ratio $k_{\text{И НЧ}}/k_{\text{И ПЛ}}$ for a ribbed plate with the measurements L=60 cm, l=5 cm, H=20 cm, h_L=h =h_1=1.2 cm. By formula (6.25) we get $k_{\text{И НЧ}}/k_{\text{И ПЛ}}$=0,256 for this structure. Therefore, the effectiveness of a vibroabsorptive coating applied to this plate, at low frequencies owing to change in its inertial-rigidity characteristics, will be less by a factor of four than for a non-ribbed plate of the same thickness h_L.

Figure 54 shows a comparison of results of calculation of $k_{\text{И НЧ}}$ for the subject ribbed plate by formula (6.24) and results of a precise calculation performed in work [92], which coincide well up to frequencies on the order of 1.0 khz.

The loss factor of a rigid vibroabsorptive coating decreases when it is applied to an I-beam in comparison to that of the same coating when applied to a plate. The loss factor of a pliable coating on the same beam stays practically the same, since the mass of the rigidity rib is less than the mass of the attached plate (see $ 25). Therefore, to damp ribbed plates at frequencies f<fpl it is advisable to use a stiffened vibroabsorptive coating or designs similar to those described in $ 31. They should be applied to the shelves of the rigidity rib, since in this case shear deformation of the viscoelastic layer, and consequently the loss factor, of these coatings will be maximum.

The formulas and recommendations set forth above hold true for a

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frequency spectrum where the condition $k_{И\ НЧ} L<1$ is satisfied. The corresponding boundary frequency f_{01}, hz, is

$$f_{01}=\frac{10^6 h_L}{4\pi^2 L^2}, \tag{6.26}$$

where h_L and L are in cm. It will be noted that $f_{01} \sim f_{p1}/3$.

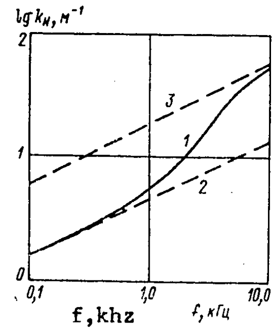

Fig. 54. Wave number of flexural oscillations in a ribbed plate.

Key: 1. precise calculattion [9]; 2. calculation by approximate formula (6.24); 3. calculation by formula (6.1) for plate with thickness h_L.

At frequencies above f_{01}, of both the rigidity rib and the plate it reinforces, flexural oscillations will occur as before, but now with different amplitude. This will take place up to the frequencies where flexural oscillations occur in the post of the rigidity rib along its thickness h_H. The corresponding boundary frequency f_{02}, hz, determined from the condition that $k_{ИН} H=1$, is equal to

$$f_{02}=\frac{10^6 h_H}{4\pi^2 H^2}, \tag{6.27}$$

where h_H and H are in cm.

In the frequency spectrum f_{01}-f_{02} the amplitude of oscillations of the space will be greater than amplitude of oscillations of the rigidity ribs because of the difference in their vibroexcitability. Taking into account also the lesser mass of the rigidity rib as compared to the mass of the plate it reinforces, it is determined

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in work [24] that in the indicated frequency spectrum less than 10% of the vibratory energy of the entire structure is contained in the rigidity ribs. However, flexural waves propagate faster in rigidity ribs than in a plate because of the former's greater rigidity. It was shown above that the ratio of velocities of flexural waves in a rigidity rib and in a plate is approximately 4. Therefore, the flow of energy in rigidity ribs can reach a value equal to approximately half the flow of energy in a plate. It is obvious that to attain a greater effect in frequency spectrum f_{01}-f_{02} it is advisable to damp both rigidity ribs and the plates they reinforce.

Thus, at frequencies lower than f_{02} it is necessary to damp rigidity ribs which reinforce plates in ship structures. A similar conclusion is drawn in work [70], which examines oscillations of ship ribbed structures at frequencies below 250 hz.

At frequencies above f_{02}rigidity ribs behave much the same as plates with thickness h_H, attached at right angles to another plate with thickness h_L. The advisability of damping rigidity ribs at the indicated frequencies is addressed in $ 23.

$ 23. Effectiveness of Damping Rigidity Ribs Which Reinforce Ship Structures

Under operational conditions application of a vibroabsorptive coating may be possible only on rigidity ribs which reinforce on structure or another. We shall evaluate the advisability of such damping at frequencies where the ribs behave much like a band which sets up lateral (flexural) oscillations along its thickness. Such oscillations, which form a diffuse field, take place in rigidity ribs at frequencies above f_{02}, determined by formula (6.27).

Assuming that at low frequencies the field of flexural waves in a plate, reinforced by the rigidity ribs, is also diffuse, an expression for a ribbed plate, according to equation (2.2), can be written as

$$W_1 - w_1\alpha_{12}c_1 + w_2\alpha_{21}c_2 - \delta_1 w_1 = 0;$$
$$w_1\alpha_{12}c_1 - w_2\alpha_{21}c_2 - \delta_2 w_2 = 0, \qquad (6.28)$$

where W_1 is vibratory energy impinging on the reinforced plate from external sources and ship structural elements joined to it; w is density of the vibratory energy; α_{12},α_{21} is the coefficient of transmission of vibratory energy from the reinforced plate to the rigidity ribs and transmission in the reverse direction; σ is the

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coefficient of absorption of vibratory energy. Index "1" applies to the reinforced plate; index "2" applies to rigidity ribs. and are determined from formulas (2.3)

Solution of equations (6.28) will give the value of t e resulting flow of energy from plate 1 into plate 2:

$$\Delta q = q_{12} - q_{21} = \alpha_{12} c_1 w_1 - \alpha_{21} c_2 w_2 = \frac{w_1 \alpha_{12} c_1}{1 + \frac{\alpha_{21} c_2}{\delta_2}}. \quad (6.29)$$

Using this expression the loss factor of the reinforced plate, which is introduced by "suction" of energy by the rigidity ribs, can, in conformity with determination of the loss factor, be derived in the form

$$\eta_p = \frac{\Delta q}{\omega w_1 S_1} = \frac{\alpha_{12} c_1}{\omega S_1 \left(1 + \frac{\alpha_{21} c_2}{\delta_2}\right)}. \quad (6.30)$$

The overall loss factor of the reinforced plate is, obviously, equal to

$$\eta_\Sigma = \eta_0 + \eta_p, \quad (6.31)$$

where η_0 is the loss factor of the reinforced plate itself.

From expression (6.30) it can be seen that the maximum loss factor value η_p under condtion that $\alpha_{21} c_1 < \delta_2$ amounts to

$$\eta_{p\,max} = \frac{\alpha_{12} c_1}{\omega S_1}. \quad (6.32)$$

From this expression it follows that the effect of damping rigidity ribs decreases with rise in frequency.

If $\alpha_{21} c_1 > \delta_2$, then from formula (6.30) it is not difficult to derive ($h_1 = h_2$):

$$\eta_p \approx \eta_2 \frac{S_2}{S_1}. \quad (6.33)$$

In this case the loss factor of the reinforced plate is proportional to the relative area of the rigidity ribs faced with a coating. An analogous result is derived for a plate partially faced with a vibroabsorptive coating (see $ 21).

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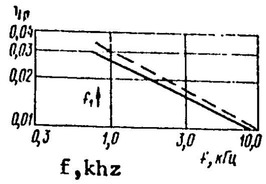

Fig. 55. Frequency characteristics ηp attributable to damping rigidity ribs of a reinforced plate.

Key: ——— ηp; ------ η_{max}

Figure 55 shows results of calculation by formula (6.30) of the loss factor ηp for an undamped steel plate with thickness h_1=1 cm, reinforced at distances of 30 cm with damped rigidity ribs with height Hp=20 cm, thickness hp=1 cm and η_2=0.2. This figure also shows the result of calculation of ηp max by formula (6.32). The value of frequency f_{02}, above which the derived results hold true, is, according to formula (6.27), equal to 850 hz. As can be seen from Figure 55, ηp comes to several hundredths and drops with rise in frequency.

It is obvious that the indicated frequencies damping of rigidity ribs, in the presence of vibroabsorptive coatings on the reinforced plate, is not advantageous, since in this case the value of η_0 will be approximately 0.1; consequently, η_0>ηp and $\eta_\Sigma \sim \eta_0$.

§ 24. The Influence of a Liquid, Contiguous to a Damped Structure, on Effectiveness of the Vibroabsorptive Coating

In a number of cases damped plates of ship hull-frame structures come in contact with liquids. Such structures include, for example, walls of fuel and other reservoirs as well as the outside of the ship's hull below the waterline. Application of vibroabsorptive coatings on wettable plates can substantially alter their effectiveness.

As is known, with flexural oscillations of a plate contiguous to a liquid, its reaction has an inertial character at frequencies below the critical frequency which is, according to data in work [24], equal to

$$f_{\text{кр}} = \frac{c_0^2 \sqrt{3\left(1-\sigma_{\text{пл}}^2\right)}}{\pi h_{\text{пл}} c_{\text{п пл}}}. \qquad (6.34)$$

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Specifically, $f_{\text{кр}}$, khz, for a metal plate contiguous to water is

$$f_{\text{кр}} \approx \frac{23}{h_{\text{пл}}}, \tag{6.35}$$

where $h_{\text{пл}}$ is in cm.

For approximation (with error on the order of 10%), this formula holds true for fuel and oils. Since the thickness of plates in ship structures usually do not exceed 2 cm, in the audio frequency band which concerns us a liquid contiguous to such a plate has the character of a connected mass.

Work [14] shows that with flexural oscillations of a plate the connected (co-oscillating) mass of liquid can be taken into account if the thickness of the layer of liquid taking part in the oscillations is taken to be $^1/_6$ of the length of the flexural wave in the plate, i.e.

$$m_c \approx \frac{\rho_0 \lambda_{\text{и}}'}{2\pi} = \frac{\rho_0}{k_{\text{и}}'}, \tag{6.36}$$

where $\lambda_{\text{и}}'$ and $k_{\text{и}}'$ refer to the plate with allowances made for its interaction with the liquid. According to [14]

$$m_c = m_{\text{пл}} \mu, \tag{6.37}$$

where μ is a function of parameters $\beta = f_{\text{кр}}/f$ and $b = \frac{\rho_0 c_{\text{п пл}}}{\sqrt{12}\,\rho_{\text{пл}} c_0}$.

Figure 56 shows the dependence of μ on β when b=0.13, which is characteristic for ship conditions. As can be seen from the figure, the co-oscillating mass increases with decrease in frequency.

The value of $k_{\text{и}}$ can be calculated by the formula

$$k_{\text{и}}' = \sqrt[4]{\frac{\omega^2 (m_{\text{пл}} + m_c)}{B_{\text{пл}}}} = \sqrt[4]{\frac{\omega^2 m_{\text{пл}} (1+\mu)}{B_{\text{пл}}}} = k_{\text{и}} \sqrt[4]{1+\mu}. \tag{6.38}$$

here $k_{\text{и}}$ referes to a plate not in contact with a liquid. Let us examine the influence of a co-oscillating liquid on effectiveness of vibroabsorptive coatings relative to a traveling flexural wave.

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As is shown in § 28, this effectiveness is determined by the index of attenuation of the amplitude of the wave

$$\gamma = \frac{1}{4} k_{\text{н}}' \eta', \qquad (6.39)$$

where k_H' and η' incorporate the influence of the liquid.

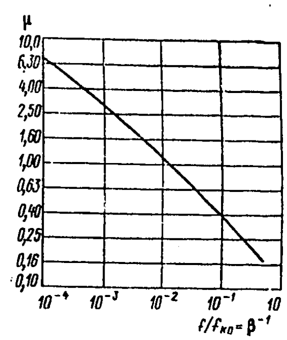

Fig. 56. Dependence of μ on $\beta^{-1}=f/f_{\text{кр}}$ for metal plates (b=0.13).

From data in Chapter 3 it follows that the loss factor of a plate with a rigid vibroabsorptive coating is determined by the elastic characteristics of the plate and elements of the coating; consequently, it is not influenced by the inertial character of a co-oscillating liquid. In this case it makes no difference on which side the liquid contacts the plate, since both sides of the plate oscillate with the same amplitude. It can therefore be considered that for a rigid coating $\eta'=\eta$.

In the case of a stiffened vibroabsorptive coating, the loss factor of which depends on the wave number of flexural oscillations in the damped plate [see formula (3.38)], the addition of liquid will cause a decrease in frequency $f_{\text{опт}}$, which according to formula (3.42) is inversely proportional to the square root of the mass of the damped plate. In this case the value of the loss factor η_{max}, when $f_{\text{опт}}$ is shifted, does not change. Thus

$$f_{\text{опт}}' = \frac{f_{\text{опт}}}{\sqrt{1+\mu}}. \qquad (6.40)$$

A shift of $f_{\text{опт}}$ toward lower frequencies causes a decrease in the loss factor at frequency $f>f_{\text{опт}}$ and an increase at frequency $f<f_{\text{опт}}$. This change in loss factor will proportional to $\sqrt{1+\mu}$.

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Nor does it make any difference here on which side the liquid comes into contact with the plate.

In case a pliable vibroabsorptive coating is used, the loss factor of the plate is inversely proportional to its mass [see formula (3.44)] and therefore for a plastic coating

$$\eta' = \frac{\eta}{1+\mu}. \qquad (6.41)$$

This assumes that the liquid comes in contact with the uncoated surface of the plate.

As a result we get:

-- for a rigid coating

$$\gamma' = \frac{1}{4} k_{\text{н}} \eta (1+\mu)^{\frac{1}{4}}; \qquad (6.42)$$

-- for a stiffened coating

$$\gamma' = \frac{1}{4} k_{\text{н}} \eta (1+\mu)^{\frac{3}{4}} \quad (f < f_{\text{опт}}); \qquad (6.43)$$

$$\gamma' = \frac{1}{4} k_{\text{н}} \eta (1+\mu)^{-\frac{1}{4}} \quad (f > f_{\text{опт}});$$

-- for a pliable coating (liquid contiguous to uncoated surface of plate)

$$\gamma' = \frac{1}{4} k_{\text{н}} \eta (1+\mu)^{-\frac{3}{4}}. \qquad (6.44)$$

From these formulas it follows that a co-oscillating liquid either increases the loss factor of a plate (stiffened coating when $f<f_{\text{ОПТ}}$ and rigid coating) or decreases it (stiffened coating when $f>f_{\text{ОПТ}}$ and pliable coating).

It is somewhat more complex with a pliable coating which is in contact with a liquid. In this case the liquid at frequencies $f<f_{\text{КР}}$ plays the role of a mass loading the free surface of the coating. The presence of such a mass causes a decrease in resonant and antiresonant frequencies of the coating layer (see § 12). Therefore, application of a pliable coating to the surface of a structure which is contiguous to a liquid serves to expand downward the frequency realm in which

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vibration damping is effective. For example, when a pliable vibroabsorptive coating (ρ_2=1 g/cm^3, c_2=3·10^4 cm/c, h_2= 2 cm) is applied to the surface of a steel plate with thickness h_1=1 cm, which is contiguous to water, the frequency of first resonance of the coating shifts from f_{p1}=3.8 khz to f_{p1}=1.8 khz (μ=0.35, μ_{32}=1.36). A precise calculation of such a situation, done by V.V. Barabanov and Yu.D. Sergeyev, showed f_{p1}=4.8 khz and f'_{p1}=2.6 khz, which agrees satisfactorily with the approximate results obtained by the simpler method.

The index of attenuation of amplitude of the flexural wave in the case of a pliable coating contiguous to a liquid is equal to

$$\gamma' = \frac{1}{4} k_{\text{и}} \eta', \tag{6.45}$$

where η' is calculated by formulas (3.52) and (3.59) with allowance made for loading of the free surface of the coating by mass m_3= m_c.

When a vibroabsorptive coating is applied to a ribbed structure, the aforesaid holds valid with the only difference being that the index of attenuation is proportional in this case to

§ 25. Damping Vibration of Beams, Pipes and other Rod Structures

In designing an antinoise system it may be necessary to damp rod-like structures. These structures are for the most part either tubes (pipes, pillars) or beams of I- or channel section (ribs and frame members). Let us examine damping of the three possible types of elastic waves in such structures: flexural, torsional and longitudinal. In all cases we shall determine the loss factor of the subject structure in the frequency spectrum where the plane section hypothesis holds true for them. At higher frequencies the oscillations of beams and tubes are much the same as oscillations of plates, and their damping is determined by formulas in Chapter 3. Limitations on applicability of the formulas given below are specified in each of the cases in question.

Damping of Tubes. With flexural oscillations of a tube with a rigid coating on its outer surface the loss factor is determined in work [34]:

$$\eta \approx \frac{\eta_2 E_2 d_2^2 h_2^2}{E_1 d_1^2 h_1^2} \approx \frac{\eta_2 E_2 h_2^2}{E_1 h_1^2}, \tag{6.46}$$

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where d_1 and d_2 are diameters of the mean surfaces of the tube and coating layer; h_1 and h_2 are thicknesses of the walls of the tube and the coating.

The loss factor in a tube with a stiffened coating, when its flexural oscillations are being damped, is calculated in work [113] and is equal to

$$\eta = \frac{\eta_2 \gamma g_2}{(1+g_2)(1+g_2+\gamma g_2)+g_2^2\eta_2^2(1+\gamma)}, \qquad (6.47)$$

where

$$\gamma = \frac{4E_3S_3}{E_1S_1} \approx \frac{4E_3h_3}{E_1h_1};$$

$$g_2 = \frac{G_2S_2}{S_3E_3k_\text{и}^2h_2^2} \approx \frac{G_2}{k_\text{и}^2E_3h_2h_3};$$

S_i is the area of the cross sections of the tube (i=1) or coating elements (i=2, 3).

There is some value $g_{2\,\text{опт}}$, at which is maximum, equal to

$$\eta_{max} = \frac{\eta_2\gamma}{2+\gamma+2g_{2\,\text{опт}}^{-1}}, \qquad (6.48)$$

where

$$g_{2\,\text{опт}} = \left[(1+\gamma)(1+\eta_2^2)\right]^{-\frac{1}{2}}.$$

When a pliable coating is applied to a flexurally-oscillating tube its loss factor can be approximately derived by formula (3.44) in which m_1 is replaced by $M_1(2d_1)^{-1}$ (M is the mass of the tube per unit of length). The maximum loss factor will be at a frequency determined by formula (3.49). The value of this maximum is calculated by formula (3.49) with the indicated substitution of $M_1(2d_1)^{-1}$ for m_1.

With longitudinal oscillations of a tube with a rigid coating its loss factor is determined by formula [34]:

$$\eta = \frac{\eta}{1+\frac{E_1S_1}{E_2S_2}} \approx \frac{\eta_2E_2h_2}{E_1h_1}. \qquad (6.49)$$

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This keeps in mind the $E_1S_1>E_2S_2$.

Damping of longitudinal oscillations of a tube by a stiffened coating can be determined by the wave mechanical resistance method (see $ 9) in the following form:

$$\eta = \frac{\eta_2 g_2 \gamma}{(1+g_2)(1+g_2+\gamma g_2)}, \qquad (6.50)$$

where
$$\gamma = \frac{S_3E_3}{S_1E_1} \approx \frac{E_3h_3}{E_1h_1};$$
$$g_2 = \frac{G_2S_2}{4E_3S_3k_n^2h_2^2} \approx \frac{G_2}{4E_3k_n^2h_2h_3};$$

k_H is the wave number of longitudinal oscillations of the tube.

In this case, when $g_2=g_{2\text{опт}}=(1+\gamma)^{-\frac{1}{2}}$ the loss factor has maximum equal to $\eta_{max}=\eta_2\gamma(2+\gamma+2g^{-1}_{2\text{опт}})^{-1}$.

The loss factor of a longitudinally-oscillating tube with a pliable coating is calculated by formula (3.44) by inserting $m_1=M_1(\pi d_1)^{-1}$ and $k_2=k_{c2}$. And as in the preceding case, the greatest value of loss factor ηp_1 here will be at the frequency of first thickness resonance of the coating, determined by formula (3.48) where k_2-k_{c2}. Factor ηp_1 is calcualted by formula (3.49) when $m_1=M_1(\pi d_1)^{-1}$ is inserted.

With torsional oscillations of a tube with a rigid coating its loss factor is determined by the wave mechanical resistance method

$$\eta = \frac{\eta_2}{1+\frac{G_1I_{p1}}{G_2I_{p2}}} \approx \frac{\eta_2G_2I_{p2}}{G_1I_{p1}} = \frac{\eta_2G_2d_2h_2\left(d_2^2+h_2^2\right)}{G_1d_1h_1\left(d_1^2+h_1^2\right)} \approx \frac{\eta_2G_2h_2}{G_1h_1}. \qquad (6.51)$$

The loss factor of a tube with a stiffened coating which undergoes torsional oscillations will be calculated by the deformation energy method (see $ 9).

According to formula (3.5) the loss factor can be written as

$$\eta = \frac{\eta_2W_{\text{пот }2}}{W_{\text{пот }1}+W_{\text{пот }2}+W_{\text{пот }3}}, \qquad (6.52)$$

where η_2 is the loss factor of the viscoelastic material; $W_{\text{пот }i}$ $(i=1,2,3)$ is the potential energy built up in the tube, the viscoelastic layer and the stiffening layer respectively. It is obvious in this case that damping is attributable to shear deformation of the viscoelastic

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layer. The values of the potential energy included in formula (6.52) should be calculated with allowances made for the deflection angle of sections of the tube and the outer layers, which takes place under torsional oscillation of the tube (Fig. 57). For the angle of deflection of sections of the tube one can write

$$\theta_1(x) = \theta_1 \sin k_\kappa x, \qquad (6.53)$$

where k_K is the torsional wave number in the tube with vibroabsorptive material applied; θ_1 is amplitude of the deflection angle.

For the angle of deflection of sections of a stiffened layer, consequently, we have

$$\theta_3(x) = \theta_3 \sin k_\kappa x. \qquad (6.54)$$

Displacement of the outer surface of the viscoelastic layer relative to its inner surface, taking (6.53) and (6.54) into account, is

$$R \cdot \theta_2(x) = R(\theta_1 - \theta_3) \sin k_\kappa x, \qquad (6.55)$$

where R is the outer radius of the section through the tube.

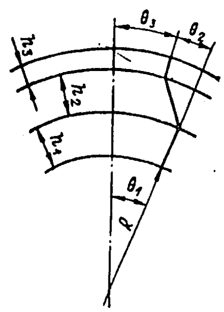

Fig. 57. Pattern of deformations of a stiffened coating on a tube subjected to torsional oscillations.

Let us calculate the potential energy $W_{\text{ПОТ}\ i}$ ($i=1,2,3$) for a piece of tube with length $l=\lambda_K/4=\pi/2k$ (λ_K is length of the torsional wave in the tube). We shall bring the left edge of this piece into coincidence with the beginning of coordinate $x=0$. The potential energy of an element of the tube with length dx, placed at a distance x from the beginning of the coordinates, is equal to

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$$dW_{\text{пот}\,1}=\frac{G_1 I_{p1}(d\theta)^2}{2dx}, \qquad (6.56)$$

where I_{p1} is the polar moment of inertia of a section through the tube; G_1 is the modulus of shear of the tube material; $d\theta$ is the angle of torsion of this element, equal to

$$d\theta=\theta_1(x+dx)-\theta_1(x)=\theta_1\frac{\pi^2(dx)^2}{4l^2}. \qquad (6.57)$$

By inserting (6.57) into (6.56) we find that

$$dW_{\text{пот}\,1}=\frac{G_1 I_{p1}\pi^2 dx}{8l^2}. \qquad (6.58)$$

From (6.58) it is not difficult to derive value

$$W_{\text{пот}\,1}=\int_0^l dW_{\text{пот}\,1}=\frac{G_1 I_{p1}\theta_1^2\pi k_{\text{к}}}{4}. \qquad (6.59)$$

To determine $W_{\text{пот}2}$ we shall isolate from the viscoelastic layer an element with volume

$$dV=dxh_2Rd\theta \qquad (6.60)$$

and determine the potential energy $dW_{\text{пот}2}$ within it.

$$dW_{\text{пот}\,2}=\frac{1}{2}G_2\varepsilon^2 dV=\frac{G_2(R\theta_2)^2\,dV}{2h_2^2}, \qquad (6.61)$$

where G_2 and h_2 are modulus of sher and thickness of the viscoelastic layer; ε is shear deformation in the isolated element when the tube is deflected at angle θ. Inserting (6.60) into (6.61) and integrating the result by θ and x, we find

$$W_{\text{пот}\,2}=\int_0^l\int_0^{2\pi} dW_{\text{пот}\,2}=\frac{\pi^2 G_2\theta_2^2 R^3}{4k_{\text{к}}h_2}. \qquad (6.62)$$

Energy $W_{\text{пот}3}$ can be found much the same as $W_{\text{пот}1}$

$$W_{\text{пот}\,3}=\frac{\pi G_3 I_{p3}\theta_3^2 k_{\text{к}}}{4}, \qquad (6.63)$$

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where G_3 is modulus of shear of the material in the stiffening layer; I_{p3} is the polar moment of inertia of a section through the stiffening layer.

It will be noted that the equivalent rigidity of the stiffening layer relative to the tangential forces is equal to

$$C_3 = \frac{j4\omega R^2}{\pi G_3 I_{p3} k_K}. \qquad (6.64)$$

The analogous value for the viscoelastic layer is

$$C_2 = \frac{j4\omega h_2 k_K}{\pi^2 G_2 R}. \qquad (6.65)$$

Using (6.64) and (6.65) through Θ_1, one can express values Θ_2 and Θ_3 in the following form, assuming that the rigidities C_2 and C_3 act in parallel

$$\Theta_2 = \frac{\Theta_1}{1+g_2};$$
$$\Theta_3 = \frac{\Theta_1 g_2}{1+g_2}, \qquad (6.66)$$

where $$g_2 = \frac{C_3}{C_2} = \frac{S_2 G_2}{S_3 G_3 k_K^2 h_2^2} \approx \frac{G_2}{G_3 k_K^2 h_2 h_3},$$

S_2 and S_3 are the areas of sections through the viscoelastic and stiffening layers respectively.

Inserting (6.59), (6.62) and (6.63) and using (6.66) we find from (6.52) the sought for value η:

$$\eta = \frac{\eta_2 g_2 \gamma}{(1+g_2)(1+g_2+g_2\gamma)}, \qquad (6.67)$$

where $$\gamma = \frac{G_3 I_{p3}}{G_1 I_{p1}} = \frac{G_3 d_3 h_3 (d_3^2 + h_3^2)}{G_1 d_1 h_1 (d_1^2 + h_1^2)} \approx \frac{G_3 h_3}{G_1 h_1}.$$

Analysis of formula (6.67) shows that at low frequencies the factor η increases in proportion to ω^2, and then having reached maximum η_{max}

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at $g2_{\text{ОПТ}}$, it begins to decrease in inverse proportion to ω^2. The value ηmax can be found by differentiating (6.67) by g2 and equating the result to zero

$$g_{2\,\text{опт}} = \frac{1}{\sqrt{1+\gamma}}. \qquad (6.68)$$

Wherein

$$\eta_{max} = \frac{\eta_2\gamma}{2+\gamma+2g_{2\,\text{опт}}^{-1}}. \qquad (6.69)$$

The loss factor under torsional oscillations of a tube, faced on its outer surface with a pliable coating, is determined in work [30]

$$\eta = \eta_2 \frac{2\,\text{sh}\,\eta_2\nu - \eta_2 \sin 2\nu}{2\,\text{sh}\,\eta_2\nu + \eta_2 \sin 2\nu + \mu_{12}\nu\eta_2(\cos 2\nu + \text{ch}\,\eta_2\nu)}, \qquad (6.70)$$

where $\mu_{12}=2\rho_1 I_{p1}/\pi\rho_2 h_2 R^3$, $\nu=k_{c2}h_2$.

Analysis of this formula shows that at low frequencies ($k_c2h2<1$) η increases with frequency:

$$\eta \approx \frac{\eta_2\nu^2}{3\left(1+\frac{\mu_{12}}{2}\right)} = \frac{\eta_2 h_2^2\omega^2}{3c_{c2}^2\left(1+\frac{\mu_{12}}{2}\right)}, \qquad (6.71)$$

where c_c2 is the velocity of the shear waves in the layer. Upon reaching the first resonance (a quarter of the wave length falls along the thickness h2), loss factor η achieves maximum value

$$\eta_{max} \approx \frac{\eta_2}{1+0,615\,\mu_{12}\eta_2^2}. \qquad (6.72)$$

With further increase in frequency η decreases, passing through maximums and minimums at resonant and antiresonant frequencies. At frequencies where $k_c2h2>1$,

$$\eta \approx 2(\mu_{12}\nu)^{-1} = 2c_{c2}(\mu_{12}\omega h_2)^{-1}.$$

The results presented fit atfrequencies where the "plane section" hypothesis holds true under flexural oscillations of a tube. In the case of longitudinal and torsional oscillations this hypothesis holds true over the entire audio frequency spectrum that concerns us.

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Under flexural oscillations the application of this hypothesis is limited by frequency [24]:

$$f_0 \approx \frac{4h_1 c_{n1}}{\sqrt{12}\,\pi d_1^2}. \tag{6.73}$$

Above this frequency the envelope properties of the tube show up and its loss factor with various types of vibroabsorptive coatings can be evaluated by the appropriate formulas from Chapter 3, assuming therein that the thickness of the damped plate is equal to the thickness of the walls of the tube h_1.

Figure 58 shows the experimentally determined amplitude of flaxural oscillations along a steel tube 1" in diameter, faced with a vibroabsorptive coating. It can be seen that at frequencies below 6.0 khz the experiment agrees well with the theory based on the "plane section" hypothesis, but at higher frequencies it agrees with representation of the tube as a plate. Calculation of the boundary frequency f_0 by formula (6.73) gives the value of 6.6 khz, which agrees well with experimental data.

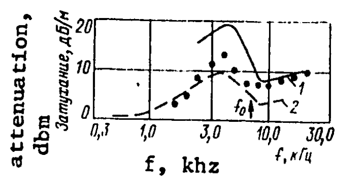

Fig. 58. Attenuation of amplitude of flexural oscillations along a damped steel tube 1" in diameter.

Key: 1. calculation for a plate with thickness the same as tube walls;
2. calculation for a rod;
● experiment

It will be noted that the value of the loss factor of a pliable vibroabsorptive coaing applied to a tube is practically independent of the value f_0, insofar as with flexural oscillations it is determined by mass of the tube and thickness of the coaing with both beam and envelope forms of oscillations. Figure 59 shows loss factors of a pliable coating with thickness 1.2 cm applied to steel tubes 1" and 6" in diameter. The character of the frequency dependence, determined by thickness of the coating, is the same on both cases, in spite of the difference in values of frequency f_0(6.6 and 0.18 khz respectively).

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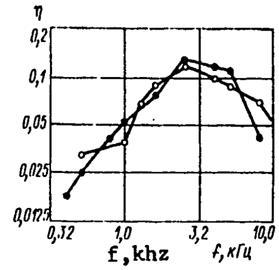

Fig. 59. Loss factor of a pliable vibroabsorptive coating 1.2 cm thick applied to a flexurally-oscillating steel tube.

Key: ●– tube 1" in diameter
O– tube 6" in diameter

Damping of Beams. Beams on ship structures are usually excited by forces which act along their walls (Fig. 60) and cause flexural oscillations within them. If a rigid vibroabsorptive coating is applied to a flexurally-oscillating beam (Fig. 60, a) the loss factor, calculated by the deformation energy method in work [113], will appear as

$$\eta = \eta_2 \frac{B_2 + D_2 h_{21}^2}{B_1 + B_2 + D_2 h_{21}^2}, \tag{6.74}$$

where B_1 is the flexural rigidity of the damped beam relative to the neutral axis (axis x in Fig. 60); $B = E_2 h_2^3 l/6$; $D_2 = 2E_2 h_2 l$; l is the width of the shelf of the beam; h_{21} is the distance of the median plane of the coating to axis x; η_2 is the loss factor of the coating material.

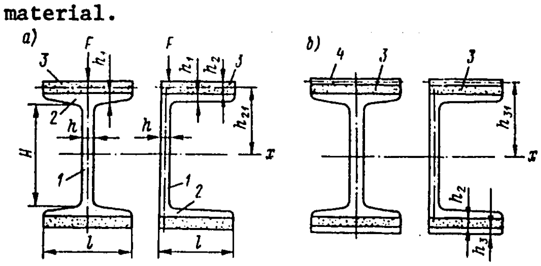

Fig. 60. Pattern of application on a beam of a rigid (a) and stiffened (b) vibroabsorptive coating.
Key: 1. beam pedestal; 2. beam shelf; 3. coating; 4. stiffener

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Usually $B_1 \gg B_2$ и $B_1 \gg D_2 h_{21}^2$. Therefore formula (6.74) can be simplified as

$$\eta \approx \eta_2 \frac{E_2 h_2 l H^2}{2E_1 I_{1x}}, \qquad (6.75)$$

where H is the height of the wall of the beams; I_{1x} is the moment of inertia of a section through the beam relative to axis x. The loss factor of a flexurally-oscillating beam with a stiffened coating (Fig. 60,b) appears as [113]

$$\eta = \frac{\eta_2 \gamma g_2}{(1+g_2)^2 + g_2^2\eta_2^2 + g_2\gamma\left[1 + g_2(1+\eta_2^2)\right]}, \qquad (6.76)$$

where $$\gamma = \frac{2E_3 h_3 h_{31}^2 l}{B_1}; \quad g_2 = \frac{G_2}{k_и^2 E_3 h_2 h_3};$$

$k_и$ is the wave number of flexural oscillations of the beam.

Analysis of formula (6.75) shows that when the value of the shear parameter g_2 is

$$g_2 = g_{2\,опт} = \frac{1}{\sqrt{(1+\gamma)(1+\eta_2^2)}}, \qquad (6.77)$$

the loss factor of a beam with a stiffened coating reaches maximum

$$\eta_{max} = \frac{\eta_2 \gamma}{2 + \gamma + 2g_{2\,опт}^{-1}}. \qquad (6.78)$$

The loss factor of a flexurally-oscillating beam, with a pliable vibroabsorptive coating applied, can be determined by formula (3.44), by replacing m_1 in it with $M_1(2l)^{-1}$ (M_1 is mass of the beam per unit of length). Just as when a pliable coating is applied to plate, the loss factor reaches maximum at a frequency determined by formula (3.49), by making the aforementioned substitution in it.

Under longitudinal oscillations of a beam with a rigid coating, according to work [34], we have

$$\eta = \frac{\eta_2}{1 + \dfrac{E_1 S_1}{2E_2 h_2 l}} \approx \frac{2\eta_2 E_2 h_2 l}{E_1 S_1}, \qquad (6.79)$$

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where S_1 is area of a section through the beam.

The loss factor of a longitudinally-oscillating beam with stiffened coating can be calculated by formula (6.50) if it assumes that

$$\gamma = \frac{2E_3h_3l}{E_1S_1}; \quad g_2 = \frac{G_2}{k_п^2 E_3 h_2 h_3}, \tag{6.80}$$

where $k_п$ is the wave number of longitudinal oscillations of the beam.

Losses in the same beam, but with a pliable coating, are calculated by formulas (3.44) and (3.49) if ml is replaced by $M_1(2l)^{-1}$ and k_2 by k_c2 (k_c2 is the wave number of the shear oscillations in the coating, which occur with tangential displacements of the damped surface of the beam.

Damping of torsional oscillations of a beam with rigid coating can be evaluated by the corresponding formulas for a tube

$$\eta = \frac{\eta_2}{1 + \dfrac{G_1 I_{p1}}{G_2 I_{p2}}} \approx \frac{\eta_2 G_2 I_{p2}}{G_1 I_{p1}} \approx \frac{\pi\eta_2 G_2 H^3 l h_2}{8G_1 I_{p1}}. \tag{6.81}$$

For a stiffened coating the loss factor of a beam which undergoes torsional oscillations can be determined by formulas (6.67) and (6.69) if they assume that

$$g_2 = \frac{G_2}{G_3 k_к^2 h_2 h_3}; \quad \gamma \approx \frac{\pi G_3 H^3 l h_3}{8G_1 I_{p1}}. \tag{6.82}$$

Finally, the loss factor for the same beam, but with a pliable coating, is calculated by formulas (6.70) and (6.72) if the following is substituted

$$\mu_{12} \approx \frac{2\rho_1 I_{p1}}{\rho_2 I_{p2}} \approx \frac{16\rho_1 I_{p1}}{\pi\rho_2 H^3 l h_2}. \tag{6.83}$$

Formulas for calculation of loss factors of damped beams are suitable in the case of longitudinal oscillations over the entire audio frequency spectrum. For torsional oscillations the indicated formulas hold true to the frequency at which flexural oscillations of the base of the beam occur; and for flexural oscillations to the frequency where flexural oscillations of the shelves of beam occur. The corresponding boundary frequencies are

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-- for an I-beam:

$$f_{0к} = \frac{10^6 h}{4H^2}, \tag{6.84}$$

$$f_{0н} = \frac{3{,}6 \cdot 10^5 h_1}{l^2}; \tag{6.85}$$

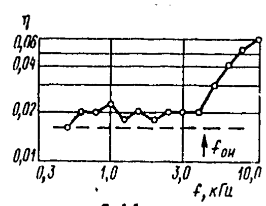
--for a channel beam

$$f_{0к} = \frac{10^6 h}{4H^2}, \tag{6.86}$$

$$f_{0н} = \frac{7{,}8 \cdot 10^5 h_1}{l^2}. \tag{6.87}$$

In these formulas h, h_1, 1 and H are in cm.

The indicated frequencies correspond to the first basic frequencies of flexural oscillations of the respective beam elements.



f,khz

Fig. 61. Loss factor of a flexurally-oscillating I-beam with a rigid vibroabsorptive coating of "Agat" plastic.

Key: 0 experiment
-- calculation by formula (6.75)

Figure 61 shows the loss factor of a steel I-beam (H=16 cm, h=hl=0.6 cm, l=7 cm), faced with "Agat" plastic (h_2=1.2 cm). Flexural oscillations were excited in the beam. From Figure 61 it can be seen that below a frequency of 4.0 khz the loss factor is low. At higher frequencies, where flexural oscillations of the separate elements of beam occur, the loss factor increases. The calculated frequency $f0_H$, determined by formula (6.85) is 4.6 khz, which corresponds well with the experiment. Figure 61 also shows the value for , calculated by formula (6.75) where $E_2/E_1 = 10^{-2}$, $\eta_2 = 0{,}25$ and $h_2 = 1{,}2$ cм.

The calculated and experimental data correspond well at frequencies below $f0_H$.

Tables 7 and 8 show results of calculation of the loss factors in a № 14 I-beam (H=14 cm, 1=8.2 cm, h=h_1=1 cm) and in a 2" diameter tube (h_1=0.4 cm, M_1=48 g) for various coatings and oscillations. Coatings with the following specifications were assumed for the calculation:

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-- rigid: η_2=0.25; $E_2=2\cdot10^{10}$ DIN/cm^2; ρ_2=1.35 g/cm^3; h_2=1 cm;

--stiffened: η_2=0.6; $G_2=10^8$ DIN/cm^2; ρ_2=1 g cm^3; h_2=0.6 cm; ρ_3=7.8 g/cm^3; h_3=0.1 cm; $E_3=2\cdot10^{12}$ DIN/cm^2;

--pliable: h_2=1.4 cm; $c_2=3.8\cdot10^4$ cm/c; ρ_2=1 g/cm^3.

The ratio of mass of the coating to mass of the damped structure was for the tube and beam 0.4 and 0.15 respectively.

Parameters of the coatings were selected on the basis of their equal mass.

Table 7

Loss Factor in a Tube 2" in Diameter

1) Покрытие	5) Колебания		
	6) Изгибные	7) Продольные	8) Крутильные
2) Жесткое	η=0,016	η=0,006	η=0,006
3) Армированное	η_{max}=0,1	η_{max}=0,034	η_{max}=0,034
4) Мягкое	η_{p1}=0,24	η_{p1}=0,3	η_{p1}=0,3

Table 8

Loss Factor in a № 14 I-Beam

1) Покрытие	5) Колебания		
	6) Изгибные	7) Продольные	8) Крутильные
2) Жесткое	η=0,003	η=0,002	η=0,003
3) Армированное	η_{max}=0,018	η_{max}=0,013	η_{max}=0,018
4) Мягкое	η_{p1}=0,15	η_{p1}=0,15	η_{p1}=0,2

Key for tables 7 and 8: 1. coating; 2. rigid; 3. stiffened; 4. pliable 5. oscillations; 6. flexural; 7. longitudinal; 8. torsional.

It can be seen from Tables 7 and 8 that the least effect is achieved when rod structures are damped with a rigid vibroabsorptive coating. Somewhat better, but practically insignificant, are the loss factors given by a stiffened coating (with the exception of damping flexural oscillations of a tube). Only the pliable vibroabsorptive coating proves effective in all cases. This advantage is explained by the fact that the effectiveness of a pliable coating is inversely proportional to the mass of the damped structure, which in tubes and beams is relatively small. At the same time the effectiveness of a stiffened, and particularly a rigid coating, depends on rigidity of the damped structure, which is very significant in tubes and beams.

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Hollow rod-like structures can be damped not only by external coatings, but by vibroabsorptive materials placed in the internal cavities of the structures. Friable vibroabsorptive materials (see $ 18) as well as viscous materials such as bitumen, which harden on cooling, may be used for this purpose. Such materials are effective for flexural oscillations of damped structures. More complex arrangements are required for other types of oscillation. For example, to damp longitudinal oscillations of tubular structures and arrangement can be used [69], which consists of a metal conductor fixed by elastic fasteners along the axis of the tube and a viscoelastic material (bitumen) poured into the opening between tube and conductor. With longitudinal oscillations of the tube the conductor does not strech, and as a result the viscoelastic material undergoes shear deformation which effectively absorbs the vibratory energy. The distance between the fastener elements must not be less than $^1/_6$ of the length of the longitudinal wave in the tube at the lowest frequency of the oscillations being damped (Fig. 62).

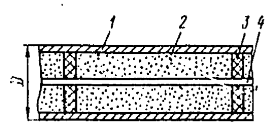

Fig. 62. Vibroabsorptive structure for damping longitudinal oscillations of a tube.

Key: 1. damped tube; 2. viscoelastic material; 3. fastening elements; 4. metallic conductor.

$ 26. Vibration Absorption in a System of Connected Spans

When a vibroabsorptive coating or other means of vibration absorption is applied to a span, which is connected with other undamped elements of a ship's hull-frame structure, the reduction of vibration in the damped span may turn out to be less that than the value described by formula (7.22). This is explained by the fact that losses of energy in the span owing to its damping will be compensated for by an influx of energy from adjacent, undamped spans. As a result the actual loss factor η_ϕ in the damped span will be somewhat less than the loss factor η in an identical, but isolated, span. We shall determine how vibration absorption in a separate span is influenced by spans connected to it by using an example of two plates. Vibratory energy W from some source (for example, from other spans or from a mechanism) arrive at one of them (denoted by index "1"). The other

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plate, denoted by index "2", corresponds to the span being examined. Let both of the plates have the same loss factor η before means of vibration absorption are applied. After application of these means on the isiolated plate 2 the loss factor within it becomes η.

Since in actual ship conditions each span is connected to several others, one can consider that the following condition is met

$$S_2 \ll S_1, \tag{6.88}$$

where S_1 and S_2 are areas of plates 1 and 2.

We shall seek out the effectiveness of means of vibration absorption, installed on plate 2, in the form

$$Э = \frac{\langle \xi_2^2 \rangle}{\langle \xi_2^2 \rangle'} = \frac{w_2}{w_2'} = \frac{q_2}{q_2'}, \tag{6.89}$$

where there is a prime ′ these are values after installation on plate 2 of means of vibration absorption. For simplicity, we shall disregard the change in velocity of flexural waves in plate 2 when the means of absorption are installed.

According to data in work [24], we have

$$q_2 = \frac{W_1 \alpha_{12} L}{\alpha_{21} L \delta_1 S_1 + \alpha_{12} L \delta_2 S_2 + \delta_1 \delta_2 S_1 S_2}, \tag{6.90}$$

where α_{12} and α_{21} are coefficients of transmission of vibratory energy from plate 1 to plate 2 and in the opposite direction; L is the perimeter of conjunction of plates 1 and 2; δ_1 and δ_2 are coefficients of absorption of energy in plates 1 and 2, equal to

$$\delta_1 = \frac{\omega \eta_0}{2c_{и1}}; \quad \delta_2 = \frac{\omega \eta_0}{2c_{и2}}. \tag{6.91}$$

Values of coefficients α_{12} and α_{21} can be determined by formulas (2.3) and (2.7) proceeding from the thickness of plates 1 and 2. In the first approximation, assuming equal thickness in the joined plates, on can consider that

$$\alpha_{12} = \alpha_{21} = \alpha_0; \quad c_{и1} = c_{и2} = c_и. \tag{6.92}$$

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Taking the aforesaid into account, by substituting formula (6.90) into expression (6.89) it is not difficult to derive the sought value of effectiveness

$$Э \approx \frac{\eta}{\eta_0} \frac{1+\gamma\left(\frac{\eta_0}{\eta}+\frac{S_2}{S_1}\right)}{1+\gamma\left(1+\frac{S_2}{S_1}\right)} = \frac{\eta_\Phi}{\eta_0}, \tag{6.93}$$

where $\gamma = \frac{2L\alpha_0 c_{\text{п}}}{\omega\eta_0 S_2}$; η_Φ is the actual loss factor of plate 2.

Taking the inequality (6.88) into account, the expression (6.93) can be re written as follows

$$Э \approx \frac{\eta}{\eta_0} \frac{1+\gamma\left(\frac{\eta_0}{\eta}+\frac{S_2}{S_1}\right)}{1+\gamma} = \frac{\eta_\Phi}{\eta_0}. \tag{6.94}$$

From expression (6.94) it can be seen that η_Φ is always less than η, since

$$\frac{\eta_0}{\eta}+\frac{S_2}{S_1}<1. \tag{6.95}$$

Let us analyze the dependence of $Э$ on γ. When $\gamma \to 0$ $\to \eta/\eta_0$, consequently, $\eta_\Phi \to \eta$. If $\gamma \to \infty$, then

$$Э \to 1+\frac{\eta S_2}{\eta_0 S_1}. \tag{6.96}$$

Accordingly

$$\eta_\Phi \to \eta\left(\frac{\eta_0}{\eta}+\frac{S_2}{S_1}\right). \tag{6.97}$$

In the last case the effectiveness of means of vibration absorption depends substantially on the ratio η/η_0 and S_2/S_1. Effectiveness will markedly differ from one if $\eta S_2 > \eta_0 S_1$. In addition, an increase in η greater than values $\eta = \eta_0 S_1/S_2$ will not result in substantial gain in η_Φ.

Figure 63 shows the qualitative dependence of the actual loss factor η_Φ in the span on the geometric parameter γ. The greastest values of η_Φ occur when $\gamma<1$ or when

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$$\alpha_0 \ll \frac{\omega\eta_0 S_2}{2Lc_n}. \quad (6.98)$$

From this inequality it can be seen that the effectiveness of damping a span, which is joined to other elements of the hull-frame, depends on conditions on the periphery of the spans, i.e. on the vibroconductivity of the span joints among themselves. The less the vibroconductivity, the more effective damping will be, since the exchange of energy between the spans will thereby be weakened. Coefficient α_0 can be reduced by the use of means of vibration insulation described in work [24]; specifically, by use of vibration-arresting masses installed around the periphery of the spans.

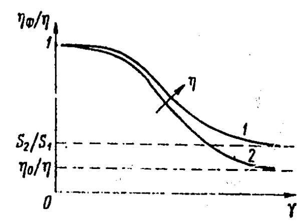

Fig. 63. Dependence of the actual loss factor of spans on γ.

Key: $1-\frac{S_2}{S_1} > \frac{\eta_0}{\eta}$; $2-\frac{\eta_0}{\eta} > \frac{S_2}{S_1}$.

$$\text{If} \quad \alpha_0 \gg \frac{S_2\omega\eta_0}{2Lc_n}, \quad (6.99)$$

then the greatest value of η_ϕ will not exceed the value of $\eta S_2/S_1$, i.e. $\eta_\phi<\eta$.

The value of α_0 can be determined by a formula taken from work [24] on the suppostion that the thicknesses of plate making up the span joints are equal

$$\alpha_0 \approx \frac{4(n-1)}{\pi^2 n^2}, \quad (6.100)$$

where n is the number of spans joined on the periphery to the damped plate. For example, for the ceiling of a cabin opening onto the upper deck, n=3.

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§ 27. Optimum Combination of Means of Vibration Absorption and Vibration Insulation

When vibroabsorptive coatings are used used together with means of vibration insulation the question of the proper combination arises.

In connection with this let is examine some examples.

Let us assume that a localized source, which excites flexural waves in a infinite isotropic plate, is surrounded by a vibration-arresting mass (VZM). The sector of the plate surrounded by the VZM is faced with a vibroabsorptive coating (VPP) (Fig. 64). Assuming the vibrational field in the surrounding part of the plate to be diffuse, the decrease in density of the energy in the plate beyond the VZM, after it is installed and the VPP applied, is described by an expression presented in work [24]:

$$\Delta = 1 + \frac{\eta\omega S}{2\alpha c_{\text{и}} L}, \qquad (6.101)$$

where η is the loss factor of the part of the plate faced with a coating; S and L are area and perimeter of the sector of the plate surrounded by the VZM; α is the coefficient of transmission of energy of lexural waves through the VZM; $c_{И}$ is the phase velocity of flexural waves in the plate.

The total mass of VZM and VPP are equal to M

$$M = M_{\text{м}} + M_{\text{п}} = \text{const}, \qquad (6.102)$$

where $M_{М}$ is the mass of VZM and $M_{П}$ is the mass of VPP.

For simplicity we shall assume VZM to be square and symmetrical relative to the neutral plane of the plate. Then $M_{М}$ will depend on the size of a side of a section of the VZM 1.

It is obvious that $M_{П}$ depends on the thickness of the coating h2. The value of 1 and h2 are linked together by the ratio of (6.102), whence

$$h_2 = \frac{M - L\rho_{\text{м}} l^2}{S\rho_{\text{п}}}, \qquad (6.103)$$

where $\rho_{М}$ and $\rho_{П}$ are density of material in VZM and VPP.

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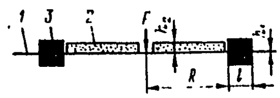

Fig. 64. Pattern of placement on a plate (1) of a vibroabsorptive coating (2) and a vibration-arresting mass (3).

Expression (6.101) can be rewritten as

$$\Delta = 1 + A\,\frac{\eta(h_2)}{\alpha(l)} = 1 + A\,\frac{\eta(l)}{\alpha(l)}. \qquad (6.104)$$

Here A is a coefficient that does not depend on mass and dimensions of the coating. Differentiating (6.104) by l and equating the result to zero, we find the condition for optimum ratio of M_M and $M_П$:

$$\frac{\eta'}{\eta} = \frac{\alpha'}{\alpha}, \qquad (6.105)$$

where the prime denotes derivative of l.

We will assume a rigid coating for which we have the loss factor (see § 10).

$$\eta \approx 13\eta_2\,\frac{E_2h_2^2}{E_1h_1^2} \equiv h_2^2, \qquad (6.106)$$

where η_2 and E_2 are loss factor and Young's modulus of the coating material; E_1 and h_1 are Young's modulus and thickness of the damped plate.
Expression (6.106) is suitable with the limits $0.1<h_2/h_1<10$. The coefficient of transmission α for the square VZM is equal to [24]:

$$\alpha \approx \frac{k_{нм}m_{пл}}{\pi k_н^2 m_м} \equiv l^{-\frac{5}{2}}, \qquad (6.107)$$

where $k_{НМ}$ and $k_Н$ are the wave numbers of flexural waves in the VZM and in the plate; $m_{ПЛ}$ is the mass of the plate falling within a unit of surface; m_M is linear mass of the VZM. Substituting (6.106) and (6.107) into (6.105) with allowances made for (6.103) we find the condition of optimum distribution of mass M between VZM and VPP for the given case

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$$\frac{M_п}{M_м} = \frac{8}{5}. \qquad (6.108)$$

A certain ratio of M_M and $M_П$ exists at which the total effect of VZM and VPP is maximum (Fig. 65). This effect is calculated at a frequency of 1.0 khz by formula (6.101) for the specific case of a steel VZM surrounding a sector of steel plate with thickness h_1=1 cm and radius R=100 cm. The coating is taken to be from plastic (ρ =1.8 g/cm^3; η_2=0.8; $E_2=2\cdot10^9$ DIN/cm^2; M=200 kg). From Fig. 65 it can be seen that with deviation from the optimum ratio of means of vibration insulation and vibration absorption, with their total mass unchanged, a decrease in effect may be very perceptible.

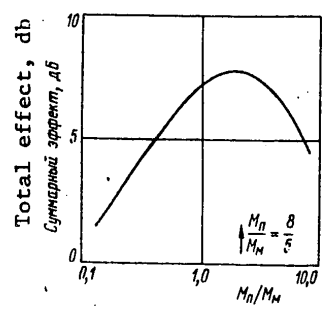

Fig. 65. Dependence of the total effect of VZM and VPP on the ratio of their mass.

Let us now examine an anologous problem for a plate, reinforced in two perpendicular directions by equidistant rigidy ribs. The vibrational field in a ribbed plate is described by expressions of heat-conductivity type equation [24]. The decrease in density of energy of flexural waves, excited in the plate by the localized source, when a ring-shaped VZM is installed at equal distance R from the source and a VPP is applied inside the encircled part of the plate, will be

$$\Delta = \frac{\gamma_0 K_1(\gamma_0 R)}{\gamma K_1(\gamma R)} \left[1 + \frac{2\alpha_0 l_0 \gamma}{\alpha} I_1(\gamma R) K_1(\gamma R)\right], \qquad (6.109)$$

where γ_0 and γ is the coefficient of vibration attenuation of the ribbed plate before and after application of VPP; $\gamma=(k_И\eta/\alpha_0 l_0)^{\frac{1}{2}}$; K_1, I_1 are cylindrical functions; α_0 is the coefficient of transmission of energy of the diffuse field of flexural waves through the rigidity

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rib ($\alpha_0 \sim 0.25$); l_0 is the distance between rigidity ribs. Expressing by formula (6.103) the loss factor η, which enters into γ, through 1, after simple calculations for the case of a square VZM, we get the following condition for optimum ratio of masses of VZM and VPP

$$\frac{\alpha'}{\alpha} \approx b \frac{\eta'}{\sqrt{\eta}}. \qquad (6.110)$$

Here

$$b = \frac{1}{2} (k_н/\alpha_0 l_0)^{\frac{1}{2}} R.$$

In deriving condition (6.110) it is assumed that $\gamma_0 R > 1$ and $\alpha_0 l_0 > \alpha R$. Substituting (6.106) and (6.107) into (6.110) with allowance made for (6.103), we get an expression for optimum size of a section of VZM

$$l^2_{опт} = \frac{5\rho_п h}{8\rho_м} \sqrt{\frac{\alpha_0 l_0 E_1}{13 k_н \eta_п E_2}}. \qquad (6.111)$$

For the example examined above $l_{опт}$=2.4 cm when l_0=30 cm. Value $l_{опт}$ does not depend on R and, consequently, is determined by the microstructure of the structure, i.e. by the distance between rigidity ribs l_0. In connection with this, for a ribbed plate the optimum ratio of $M_М$ and $M_П$ depends on their overall mass M.

$$\frac{M_п}{M_м} = \frac{M}{2\pi R l^2_{опт}} - 1, \qquad (6.112)$$

where $l_{опт}$ is determined by formula (6.111).

For other means of vibration insulation (for example, vibration insulating shock absorbers) the optimum ratios of their mass to the mass of the vibroabsorptive coating will be different.

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Chapter 7. PRACTICAL USE OF MEANS OF VIBRATION ABSORPTION ON SHIPS

§ 28. Methods of Evaluating the Effectiveness of Vibration Absorption in Ship Structures

Means of vibration absorption in ships are recommended for use to:

-- increase attenuation of vibrations which propagate through structures which link a source of vibration with sound-radiating enclosures, by application of vibroabsorptive coatings to these structures;

-- decrease sound radiation of enclosures, by damping their vibrations.

We shall examine possible methods of evaluating the effectiveness of means of vibration absorption in the indicated variants. The simplest realization of the first variant will be an infinite uniform plate, on which a source of flexural waves is installed. The sector of the plate's surface situated at a distance l from the source, represents a sound-radiating enclosure.

We shall assume that in one case sonic vibrations in the form of a flat flexural wave (the unidimensional case) propagates through this plate; in the second case the vibration is in the form of a cylindrical flexural wave (the bidimensional case). The first case corresponds to application of a coating at a distance away from the source of vibration (mechanism); the second -- in direct proximity to the source (around it). It will be noted that the first case corresponds also to propagation of a flexural wave along a rod-like structure.

We shall consider losses of oscillatory energy in the plate faced with a coating as a complex representation of the wave number of flexural waves in the plate [24]:

$$k_{H}=k_{H0}\left(1-j\frac{\eta}{4}\right), \qquad (7.1)$$

where η is the loss factor of the plate; k_{H0} is the wave number modulus.

Distribution of the amplitude of the flat flexural wave, traveling along coordinate x, is described by the expression

$$\xi(x)=\xi(0)\,e^{-jk_{H}x}, \qquad (7.2)$$

where ξ is the amplitude of the lateral displacement of the plate.

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Substituting (7.1) into (7.2) we find

$$\xi(x) = \xi(0)\, e^{-jk_{\text{н}0}x} e^{\gamma x}, \qquad (7.3)$$

where $\gamma = k_{И}\upsilon\eta/4$ is the index of amplitude attenuation.

From (7.3) it can be seen that in the presence of losses in the plate the amplitude of the flat flexural wave traveling along this plate decreases on an exponential curve away from the source of the wave.

We shall determine the effectiveness of the coating as a ratio of amplitude of vibration of the plate at point x=1 without a coating applied to the amplitude with the coating

$$Э = \frac{\xi(l)}{\xi'(l)} = e^{k_{\text{н}0}\frac{\eta}{4}l}, \qquad (7.4)$$

or in decibels

$$Э = 2{,}15 k_{\text{н}0} \Delta\eta l, \qquad (7.5)$$

where $\Delta\eta = \eta' - \eta$. The prime here denotes presence of a coating on the plate. The quantitiy without the prime refers to the plate without a coating.

Distribution of amplitude of the cylindrical wave traveling away from the source is

$$\xi(r) = \xi(0)\, H_0^{(2)}(k_{\text{н}}r), \qquad (7.6)$$

where $H_0^{(2)}$ is a Hankel Function of the second kind.

At sufficiently great distances from the source ($k_{И}r > 1$) expression (7.6) may be replaced by its asymptotic presentation

$$\xi(r) \approx \xi(0) \sqrt{\frac{2}{\pi k_{\text{н}} r}}\, e^{-i\left(k_{\text{н}}r - \frac{\pi}{4}\right)}. \qquad (7.7)$$

It will be noted that at frequency 0.1 khz when $h_{ПЛ}=1$ cm the condition $k_{И}r>1$ is satisfied, beginning from $r=r_0=17$ cm. At higher frequencies this distance is still less.

If the coating is applied around the source at distance R=1, then effectiveness of the coating, taking expression (7.6) into account, will be

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$$Э = \frac{\xi(R)}{\xi'(R)} = \frac{H_0^{(2)}(k_и l)}{H_0^{(2)}(k'_и l)}, \qquad (7.8)$$

where $k'_и$ is the wave number of the coated plate.

Substituting expressions (7.7) and (7.1) into (7.8) we get

$$Э \approx \sqrt{\frac{1 - j\frac{\eta}{4}}{1 - j\frac{\eta'}{4}}} \, e^{k_{и0}\frac{\Delta\eta}{4}l} \approx e^{k_{и0}\frac{\Delta\eta}{4}l}. \qquad (7.9)$$

This takes into account that in practice $\eta'/4<1$ and $l>r$.

Comparison of (7.9) and (7.4) shows that effectiveness of the coating is determined by the length of the coated part of a uniform plate, independent of spatial characteristics of the flexural wave traveling through the plate.

For a ribbed plate at frequencies where the field of flexural waves has a diffuse character it is customary to determine vibration by density of energy w. In this case the ratios analogous to expressions (7.2) and (7.6) appear as [24]:

-- for the unidimensional case

$$w(x) = w(0)\, e^{-\gamma x}; \qquad (7.10)$$

-- for the bidimensional case

$$w(r) = w(0)\, K_0(\gamma r), \qquad (7.11)$$

where K_0 is a cylindrical function; γ is the coefficient of vibration attenuation, equal to

$$\gamma = \sqrt{\frac{k_и \eta}{\alpha_0 l_0}}, \qquad (7.12)$$

α_0 is the coefficient of transmission of energy of the diffuse field of flexural waves through the rigidity rib (α_0=0.25); l_0 is the distance between rigidity ribs.

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When $k_{\text{И}}r>1$ expression (7.11) can be replaced by its asymptotic expresion

$$w(r) \approx w(0) \sqrt{\frac{\pi}{2\gamma r}} e^{-\gamma r}. \tag{7.13}$$

Taking into account that $w=m\omega^2\xi^2/2$, the effectiveness of a vibro-absorptive coating applied to a ribbed plate will be:

-- for the unidimensional case

$$Э=\left[\frac{w(l)}{w'(l)}\right]^{\frac{1}{2}}=e^{\frac{\gamma'-\gamma}{2}l}; \tag{7.14}$$

-- for the bidimensional case

$$Э=\left[\frac{w(l)}{w'(l)}\right]^{\frac{1}{2}}=\left[\frac{K_0(\gamma l)}{K_0(\gamma' l)}\right]^{\frac{1}{2}}. \tag{7.15}$$

When $r=r_0$ ($k_{\text{И}}r_0=1$) expression (7.15) according to formula (7.13) assumes the form

$$Э\approx \sqrt[4]{\frac{\gamma'}{\gamma}}e^{\frac{\gamma'-\gamma}{2}l}=\sqrt[8]{\frac{\eta'}{\eta}}e^{\frac{\gamma'-\gamma}{2}l}. \tag{7.16}$$

Taking into account that in practice η' usually exceeds η by an approximate factor of 10, then with error of less than 1 db, from expression (7.16) it follows that

$$Э\approx e^{\frac{\gamma'-\gamma}{2}l}. \tag{7.17}$$

Comparison of formulas (7.14) and (7.17) shows that in the case of a ribbed plate the effectiveness of a vibroabsorptive coating is also independent of the spatial characteristics of the field of flexural waves and is, in db, equal to

$$Э\approx 4{,}3(\gamma'-\gamma)l=4{,}3l\sqrt{\frac{k_{\text{и}}}{\alpha_0 l_0}}\left(\sqrt{\eta'}-\sqrt{\eta}\right). \tag{7.18}$$

From expression (7.5) and (7.18) it follows that an increase in effectiveness of a vibroabsorptive coating, relative to vibration

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propagating through ship structures, can be achieved by using a coating with a high loss factor η and lengthening the damped part of the structure l.

The indicated expressions were derived on the supposition that application of a coating to plates of a structure does not alter the wave number of flexural oscillations $k_И$ in them. This number is equal to

$$k_{и} = \sqrt[4]{\frac{\omega^2 m}{B}} \qquad (7.19)$$

and depends, consequently, on the mass of the plate falling within one unit of surface m and flexural rigidity of the plate B. Application of a coating on a plate somewhat increaes its mass and rigidity. Taking the changes in these characteristics into account, expressions (7.5) and (7.18) can be derived as

-- for a uniform plate

$$\text{Э} = 2{,}15 k_{и} l \eta \left(\frac{\eta'}{\eta} \sqrt[4]{\frac{m'B}{mB'}} - 1 \right), \qquad (7.20)$$

-- for a ribbed plate

$$\text{Э} = 4{,}3 l \sqrt{\frac{k_{и}\eta'}{\alpha_0 l_0}} \left(\sqrt{\frac{\eta'}{\eta}} \sqrt[8]{\frac{m'B}{mB'}} - 1 \right). \qquad (7.21)$$

Here $m' = m_{ПЛ} + m_{П}$ (m is the mass of the coating). The flexural rigidity of a plate with a cotaing can be calculated by formula (3.16).

From formulas (7.20) and (7.21) it can be seen that an increase in the mass of the plate results in an increase in effectiveness of the coating, while an increase in rigidity results in a decrease of effectiveness. This is explained by the fact that an increase in the mass of the plate shortens the length of the flexural wave within it, while an increase in rigidity, to the contrary, lengthens it. Therefore, given the same loss factor and mass for different plates, pliabel vibroabsorptive coatings are preferable, since they cause no practical change in rigidity of the damped plate.

Effectiveness of a vibroabsorptive coating applied directly to sound radiating enclosures can be determined by the decrease of the mean square of the amplitude of their vibrations. For flexurally-oscillating enclosures this effectiveness Э, in db, is determined in work [48]

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$$Э = 10 \lg \frac{<\xi^2>}{<\xi^2>'} = 10 \lg \frac{\eta'}{\eta} \sqrt{\frac{m'B}{mB'}}. \quad (7.22)$$

It is kept in mind here that excitation of a plate occurs along its periphery at a given oscillatory velocity at the edges in a band of frequencies which attenuate several frequencies of basic oscillations of the plate. The character of the influence of change in mass and rigidity of the plate on effectiveness of the applied vibroabsorptive coating is the same in this case as in the preceding variant.

According to work [48], with excitation in the plate of a discrete mode on the basic frequency the effectiveness of damping Э, in db, of its oscillations is equal to

$$Э = 10 \lg \frac{<\xi^2>}{<\xi^2>'} = 20 \lg \frac{\eta'}{\eta}, \quad (7.23)$$

where amplitude of the oscillatory velocity is also assumed to be fixed along the periphery of the plate. A discrete mode of oscillations of the plate can be damped more effectively than wideband excitations of the same plate [34]. Damping of a single mode of flexural oscillations of a plate does not depend on changes in its mass and rigidity when the coating is applied.

Formulas (7.22) and (7.23), which correspond to constant oscillatory velocity on the periphery of the plate, give somewhat understated values for effectiveness of a coating. As a matter of fact, damping of a single enclosure in ship structures, due to its existing oscillatory link with adjacent enclosures, will cause some decrease in amplitude of their vibrations and a corresponding decrease in vibrations of the periphery of the damped enclosure.

The formulas presented above for evaluating effectiveness of a vibroabsorptive coating relative to vibrations which propagate through ship structures are, strictly speaking, applicable when the source of vibration and sound-radiating span are linked by only one structure. If there are several such structures, then the vibratory energy arrives at the sound-radiating enclosure from the source by several paths. Evaluation of the effectiveness of vibroabsorptive coatings by formulas (7.5) and (7.18) is practically impossible, since with their use one cannot determine the ratio of vibratory energies arriving at the plate in question by the various paths. The indicated evaluation can be performed in this case, with accuracy sufficient for practice, by the Westphal method, which is based on the supposition that the vibrational fields are of a diffuse nature in the separate elements of complex engineering structures. Such an evaluation can be

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performed either with the aid of an analog electrical model of the ship's hull-frame according to work [34], or by calculation using a computer, as shown in work [8].

Evaluation of effectiveness of vibroabsorptive coatings, applied to ship structures, with the aid of an analog model is more convenient and more graphic. By changing the elements of this model according to the pattern of application of the coating, it is not difficult to determine the effectiveness of various patterns and select the best one from the point of view of acoustics, weight and cost.

The formulas presented for evaluation of effectiveness of vibroabsorptive coatings do no take into account such factors as the presence of other types of waves in addition to flexural waves: first of all, longitudinal waves, the influence of a liquid contiguous to plates, through which waves propagate and, finally, the influence of nonresonant radiations of a flexurally-oscillating plate.

The noted factors may be taken into account as follows.

Influence of Longitudinal Waves. Mechanisms excite mainly flexural waves in structures. The energy of these waves is partially converted into longitudinal waves on the path of their propagation through the structures due to the presence of various obstructions (rib, plate joints, etc.). According to data in work [24], in the first approximation the flow of energy from flexural and longitudinal waves in ship structures is equal, evidenced by the intense reciprocal conversion of these waves. The influence of longitudinal waves on the effectiveness of vibroabsorptive coatings must be substantial in ribbed plates, where obstructions for flexural waves are spaced very close. A rough evaluation of this influence in this case can be performed on the supposition of weak damping of longitudinal waves and unobstructed passage of these waves through the rigidity ribs. With this purpose in mind we shall isolate in a ribbed panel a space i with currents of energy of flexural and longitudinal waves $q_{Иi}$ and $q_{Пi}$, with $q_{Иi}=q_{Пi}$. Energy arrives at space i+1 somewhat weakened due to the effectiveness $Э_И(l_0)$ of the vibroabsorptive coating applied on the ribbed plate (l_0 is the distance between rigidity ribs). Energy $q_{Пi}$ arrives at space i+1 without any weakening. Both components of the energy in space i+1 are equally divided between flexural and longitudinal waves. Thus the flow of energy of flexural waves in space i+1 will be equal to

$$q_{Иi+1} \approx \frac{q_{Пi}}{2Э_И(l_0)} + \frac{q_{Пi}}{2}. \qquad (7.24)$$

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The sought effectiveness of the vibroabsorptive coating on a ribbed plate, taking into account the influence of longitudinal waves, will be

$$Э_{нп}(l_0)=\frac{q_{нl}}{q_{нl+1}}\approx\frac{q_{пl}}{\frac{q_{пl}}{2Э_{п}(l_0)}+\frac{q_{пl}}{2}}=\frac{2}{1+\frac{1}{Э_{п}(l_0)}}. \quad (7.25)$$

From formula (7.25) it follows that the value of maximum effectiveness of the vibroabsorptive coating, allowing for the influence of longitudinal waves, is approximately 2-3 db per space of the ship's hull.

The Influence of a Contiguous Liquid. The effectiveness of a vibroabsorptive coating, as follows from formula (7.5) and (7.8), depends on the wave number of flexural oscillations of the plate $k_И$. In the presence of a liquid contiguous to the plate the length of the flexural wave within it decreases and, consequently, the effectiveness of the coating also increases due to the additional mass. The wave number, with allowance made for influence of the liquid, can be determined by formula (6.38).

The Influence of Non-Resonant Radiation of a Flexurally-Oscillating Plate. A force acting on a plate excites within it both resonant (frequency of force ω_0 coincides with the basic frequency of the mode ω_i), and nonresonant ($\omega_0>\omega_i$ or $\omega_0<\omega_i$) modes of flexural oscillation. Resonant modes have a greater amplitude of oscillations than do nonresonant modes. However, due to better radiation capacities nonresonant low-frequency ($\omega_i<\omega_0$) modes can make a substantial contribution to the total sound radiation of the plate. The amplitude of oscillations of nonresonant modes does not depend on the dissipative properties of the plate. Therefore, with an increase in the loss factor of the plate and the corresponding reduction in amplitude of oscillations of resonant modes, the radiation of nonresonant modes limits the effect of damping sound-radiating enclosures.

The effectiveness of damping enclosures, relative to noise that they radiate, can be approximately evaluated by data in work [32]

$$Э=20\lg\frac{p}{p'}=10\lg\frac{1+A(\eta)}{1+A(\eta')}, \quad (7.26)$$

where p is the sonic pressure of radiated air noise;

$$A=\frac{\mu_\omega}{2\eta}\left[\pi-\mathrm{arctg}\frac{\eta\mu_\omega}{1-\left(1+\frac{\eta^2}{2}\right)\mu_\omega^2}\right],$$

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$\mu_\omega = \frac{\omega_1}{\omega}$, ω_1 is the frequency of the first mode of flexural oscillations of the plate.

Figure 66 shows the dependence of Э, calculated by formula (7.26) with an increase in loss factor in an enclosure from η=0.01 to η'=0.1. It also shows the effectiveness of damping relative to vibration of the same enclosure, determined by formula (7.22). Effectiveness of damping the enclosure relative to radiated noise is less than that relative to vibration. The influence of nonresonant modes is more substantial the higher the frequency. When $\mu\bar{\omega}^1$=100 effectiveness against vibration in the subject case amounts to 10 db, but against noise a total of 3.4 db.

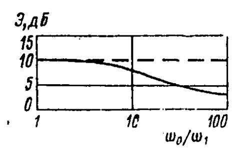

Fig 66. Effectiveness of damping an enclosure against noise (——) and vibration (---) depending on the ratio of frequency of excitation ω_0 and the first basic frequency of flexural oscillations of the enclosure ω_1.

The indicated influence of nonresonant radiation of damped plates is one of the reasons for the difference in effectiveness of damping sound-radiating enclosures relative to vibration and noise.

§ 29. The Effectiveness of Various Patterns of Application of Vibroabsorptive Coatings on Ship Structures

The comparative effectiveness of various patterns of damping of ship structures requires that an appropriate experiment for each pattern be conducted on one and the same ship. However, it is difficult to conduct such an experiment in practice. It is not advantageous to compare the effectiveness of various patterns of damping derived from differenct ships for the purpose of determining the best variants, because of possible differences in conditions of the experiments.

A solution to the problem may be measurements made with the aid of an analog electrical model of a ship's hull-frame, which permits easy determination of effectiveness in any pattern of application of a vibroabsorptive coating. Basic results of such an experiment, described in work [34], are shown below. The principle on which the model was built is expalined in the same work.

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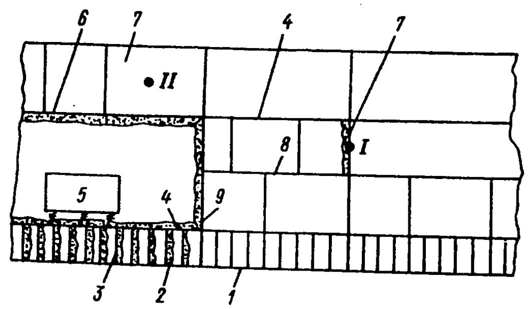

Fig. 67. Part of a ship's hull-frame and pattern of placement on it of vibroabsorptive coatings.

Key: 1. keel; 2. vibroabsorptive coating; 3. risers; 4. second bottom; 5. main engine; 6. ceiling (second deck); 7. cabin bulkhead; 8. first deck; 9. engine room (MO) bulkhead.

An analog electrical model was built for part of the hull-frame of a ship with a 900-ton displacement (Fig. 67). Using this model, the effectiveness of various patterns of application of vibroabsorptive coatings was studied. Measurements were made for different variants of use of vibroabsorptive coatings (Table 9). It was assumed that the loss factor of a hull-frame structure increases by a factor of 10 with application of a vibroabsorptive coating. The pattern of application of the vibroabsorptive coating is shown in Fig. 67. The effectiveness of the different variants of use of vibroabsorptive coatings was determined by the reduction in levels of sonic vibrations in compartments situated toward the bow from the engine room (MO) (point I, Fig. 67) and over the MO (point II, Fig. 67). It is assumed that the main engine, installed in MO, is running. Results of measurements at points I and II are shown in Table 10 in the form of effectiveness values at frequencies of 0.1, 1.0 and 10.0 khz, as well as average values over the indicated frequency spectrum. Examination of these data shows the following.

Application of a vibroabsorptive coating on the second bottom in the MO, where the operating machinery is located, reduces the levels of vibration over the entire hull-fram of the ship an average of 6 db. Application of a coating to stringers and risers in the MO adds a total of 2 db to this effect in compartments located apart from the MO. This does not alter the level of vibration in compartments over the MO, since the main part of vibratory energy from the primary engine arrives at these regions of the ship's frame through the MO

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bulkheads and side walls. For the same reason, application of a coating on the bulkheads, side walls and ceiling of the MO reduces sonic vibration at point II by an average of 10 db, but at point I a total of 3 db. Thus coating the bulkheads, side walls and ceiling of the MO achieves the greatest effect in compartments situated over the MO. The main part of the effect (7.3 db) is herein attributable to coating of the ceiling of the MO, since in this case the MO ceiling represents an enclosure (deck) of the subject cabin. Application of a vibroabsorptive coating directly to a span, where sonic vibrations must be reduced, gives a localized effect, equal on the average to 4.3 db.

Coating the hull-frame of a ship according to the complete pattern for this example (variant 7) gives an effect over the entire ship equal on the average to 10-15 db, with this effect reaching 20 db at 10 khz. In this case the best results are achieved in compartments located over the MO.

Table 9

Variants of patterns of application of vibroabsorptive coatings on a ship

Variant	Place where vibroabsorptive coating is applied
1	Floor of second bottom in MO
2	Stringers and risers between second bottom and keel in MO
3	Variant 1 plus variant 2
4	Bulkheads and side walls in MO
5	Ceiling in MO
6	Variant 4 plus variant 5
7	Variant 3 plus variant 6
8	Bulkhead in cabin

The example presented gives a graphic presentation of the possibilities of using an electrical model of a ship's hull-frame for optimum placement of means of vibration absorption on a ship.

The results obtained by measurements in the electrical model agree satisfactorily with measurements made on ships.

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Table 10

Effectiveness of Various Patterns of Application of Vibroabsorptive Coatings on Ships

Вариант схемы по табл. 9 (1)	(2) Эффективность схемы, дБ			
	на частоте 0,1 кГц (3)	на частоте 1,0 кГц (3)	на частоте 10,0 кГц (3)	среднее значение в диапазоне 0,1—10,0 кГц (4)
1	4/4	6/6	8/8	6/6
2	0/2	0/2	0/2	0/2
3	4/6	6/8	8/10	6/8
4	1/2	2/3	3/4	2/3
5	5/0	8/0	9/0	7,3/0
6	6/2	10/3	12/4	9,3/3
7	10/8	16/10	20/12	15,3/10
8	0/3	0/4	0/6	0/4,3

Figure in numerator defines reduction in vibration at point II;
Figure in denominator defines reduction at point I (see Fig. 67)

Key: 1. pattern variant from Table 9; 2. effectiveness of the pattern, db; 3. at frequency __ in khz; 4. average value in the spectrum __ khz.

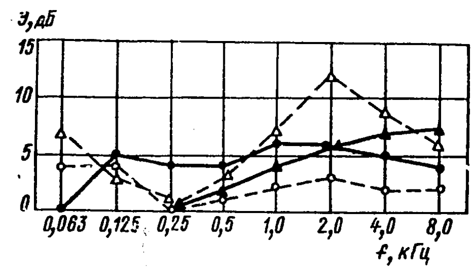

Fig. 68. Reduction of noise levels in ship compartments by use of vibroabsorptive coatings.

Key: Δ – in cabin of automobile ferry (coating on half of cabin enclosure) [112];
O– in cabin of automobile ferry (coating on one of cabin enclosures) [112];
▲ – in compartment of river boat (data of G.D. Izak);
● – in salon of motorship "Raketa" (coating in MO area).

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Figure 68 shows frequency characteristics of the effectiveness of applying a vibroabsorptive coating on various ships. Application of a coating on enclosures of a cabin on an automobile ferry reduced the noise in it an average of 5.7 db [112]. This agrees well with variant 8 (Table 10), where an average of 4.3 db was achieved. According to data in work [112], application of a coating on only one wall of the cabin gave a total effect of 2 db.

Application of a vibroabsorptive coating on the structures surrounding the MO in a "Raketa" motorship reduced the noise in its salon by an average of 4.9 db [8]. According to data of G.D. Izak, a similar application of a coating on a river boat reduced the noise in its compartment an average of 3.7 db. Work [116] shows results of using vibroabsorptive coatings on the foundations of the main mechanisms, the floor of the second bottom and the side walls of the MO. The effect of this measure ranged 6-8 db. According to data of B.D. Tartakovskiy and V.B. Chernyshev, application of a vibroabsorptive coating on structures in the area of the MO on a "Kometa" motorship reduced noise in its compartments an average of 5-6db. The results agree satisfactorily with the effectiveness of variants 1 and 4 of coating application patterns, derived by the use of an analog ship model (see Table 10).

Thus depending on the pattern of application of vibroabsorptive coatings, the noise levels in ship compartments can be reduced by 15 db and more. Work [93] refers to higher effectiveness values for vibroabsorptive coatings (10-20 db) under ship condition.

§ 30. Principles of Rational Use of Vibroabsorptive Coatings on Ships

Vibroabsorptive coatings are used on ships as the primary means of vibration absorption; therefore, we should deal in more detail with the principles of the rational use of these coatings on ships. It should be noted that the principles set forth below, in the main, hold true for other means of transportation (rail transport, aircraft, automobiles, etc).

The amount of vibratory energy in plates, when vibroabsorptive coatings are applied to them, is proportional to the amplitude of oscillations of the damped plates. Therefore, the effectiveness of one and the same amount of coating will be more significant as the coating is placed closer to the source of vibrations, where vibration of structures have the highest amplitude.

The second basic principle of rational use of vibroabsorptive coatings is the necessity that a coating be applied on all path of transmission of vibratory energy from the source to the point of observation.

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Herein the comparative quantity of vibration attenuation over the length of such paths must be taken into account. Where the difference in attenuation on the examined and shortest paths exceeds the effect expected from application of a coating, it is not advantageous to apply a coating. Figure 69 shows examples of correct and incorrect use of vibroabsorptive coatings.

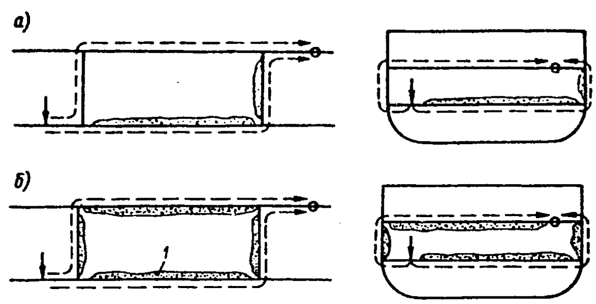

Fig. 69. Examples of patterns of application of vibroabsorptive coatings on ship structures: a- incorrect variant; b- correct variant.
---→ paths of propagation of vibrations from source to point of observation 0.
1. coating.

Non-compliance with the principle of damping all paths of vibration propagation can result in a reduced effect of the pattern of use of the coating as a whole, as was the case, for example, in an instance described in work [63].

When vibroabsorptive coatings are used within the confines of one compartment they should be applied first of all to the enclosure with the highest amplitudes of vibration. If the amplitude of vibration of the enclosure is less that the greatest values for a given compartment by a quantity which exceeds the expected effect of a coating, then it is not advantageous to damp this enclosure.

Selection of a type of vibroabsorptive coating must take into account the character os the spectrum of vibrations of the damped structures and the peculiarities of the latter.

Figure 70 shows frequency characteristics of the loss factor of a steel plate with thickness h_1=0.6 cm with various vibroabsorptive coatings. Calculation is done by the appropriate formulas from Chapter 3. The relative mass of the coatings in all cases is assumed to be 25% of the mass of the damped plate.

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For the calculations coatings with the following specifications were selected:

-- rigid: $\eta_2=0.25$; $E_2=2\cdot10^{10}$ DIN/cm^2; $\rho_2=1.35$ g/cm^3; $h_2=0.9$ cm ("Agat").

-- rigid: (from "Agat" plastic) with an intermediate layer (of foam plastic PCV-1) $\eta_3=0.25$; $E_3=2\cdot10^{10}$ DIN/cm^2; $\rho_3=1.35$ g/cm^3; $h_3=0.5$ cm; $h_2=5$ cm; $G_2=4\cdot10^8$ DIN/cm^2; $\rho_2=0.1$ g/cm^3.

--stiffened: $\eta_2=0.6$; $\rho_2=1$ g/cm^3; $G_2=10^8$ DIN/cm^2; $h_2=0.7$ cm; $\rho_3=1.7$ g/cm^3; $E_3=10^{11}$ DIN/cm^2; $h_3=0.3$ cm.

--pliable: $h_2=1.2$ cm; $\rho_2=1$ g/cm^3; $c_2=3.8\cdot10^4$ cm/c.

In addition to the loss factors, the characteristics frequencies for the listed coatings were calcualted from formula in Chapter 3. For the rigid coating and the rigid coating with intermediate layer frequencies were determined, above which effectiveness of the coatings drops due to shear deformations in the coating. These frequencies are $f_1=10$ khz and $f_1=2$ khz respectively. For stiffened and pliable coatings frequencies with maximum loss factor were determined, equal to $f_{опт}=0.67$ khz and $f_{p1}=8$ khz respectively.

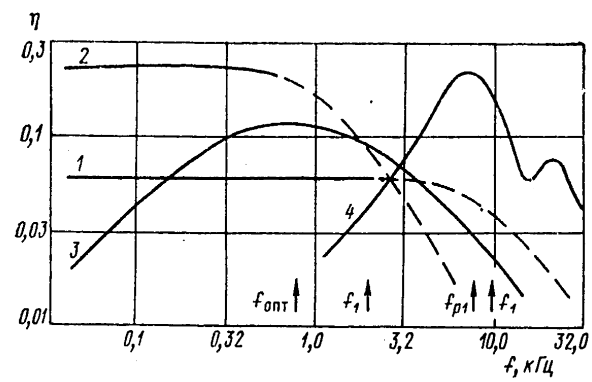

Fig. 70. Loss factor of a steel plate 0.6 cm thick with application of various types of vibroabsorptive coatings.

Key: 1. rigid coating; 2. rigid coating with intermediate layer; 3. stiffened coating; 4. pliable coating

From examination of Fig. 70 it follows that when it is necessary to damp vibrations of elastic structures (foundations, decks, bulkheads, etc.) the rigid coating with intermediate layer is preferable at low frequencies. If it is necessary to combat vibration over the entire audio frequency spectrum, the rigid coating will be best. If a high

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level of vibration is present in the middle frequency range a stiffened coating can be used. At high frequencies the best results are achieved by using a pliable coating.

The frequency spectrum can be expanded and loss factors at given frequencies increased by using combination coatings (see § 13).

On rod-like structures (pipes, pillars, etc) the best results can be acheived by use of pliable coatings. This holds true for both the frequency spectrum of beam forms of oscillation of such structures and for higher frequencies.

The main condition is placement of the coating on a damped plate. It is recommended that a rigid coating be applied to one side of a plate. This ensures high loss factor values over the entire frequency spectrum. A pliable coating is also best applied to one side of a plate. This serves to extend the frequency spectrum of effective damping into lower frequencies than application of the same amount of coating on two sides. If the damped plate is in contact with a liquid, it is preferable that the pliable coating be applied to the wettable surface. It goes without saying that the material used must be resistant to the liquid with which it comes into contact.

The thickness of a vibroabsorptive coating should be selected proceeding from the existing mass, taking into account insurance of optimum combination of thickness and length of coating. Selection of thickness and length of a coating on damped structures must depend on the area of application (near the source of vibration or remote from it) and the type of structure (uniform plate or ribbed plate). In this selction the thickness of a rigid coating or rigid coating with intermediate layer must not exceed values dictated by the absence of shear deformations in their layers over the entire frequency spectrum where effectiveness of the coatings is required (see § 13).

The principles of otimum placement of vibroabsorptive coatings on ribbed plates have their peculiarities. If it is necessary to damp a ribbed plate at frequencies lower than $f_{\Pi Л1}$ (first resonant frequency of flexural oscillations of a sector of plate bounded by adjacent rigidity ribs), at which the ribbed plate oscillates as an orthotropal plate, rigid and stiffened coatings should be applied, first of all on the shelves of the rigidity ribs, where tangential displacements of the structure under flexure are the greatest [89]. A pliable coating, the useful effect of which is attributable to lateral oscillations of the damped plate, under these conditions should be applied to the plate itself.

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The best damping of a ribbed plate at frequencies above $f_{\Pi\Pi 1}$ can be achieved by applying vibroabsorptive coatings of any type directly to the plate, since at these frequencies the amplitudes of vibrations of the plate are substantially greater than amplitudes of vibrations of rigidity ribs [8]. Frequency $f_{\Pi\Pi 1}$ for actual ship structures is approximately 0.1-0.3 khz.

In those cases when for some reason a vibroabsorptive coating cannot be applied to the plate, some effect at high frequencies can be derived by damping the rigidity ribs. When coating is applied to a reinforced, applying it to the ribs as well is ineffective at the indicated frequencies.

When vibroabsorptive coatings are used in conjunction with means of vibration insulation one should keep in mind the advantage of such an optimum distribution of the existing weight between them at which their combined effect will be greatest. Additional information on the optimum use of vibroabsorptive coatings can be found in works [46, 48, 110].

§ 31. Recommendations for Use of Means of Vibration Absorption on Ships

General recommendations for the use of means of vibration absorption on ships can be grouped under three types of use patterns.

In the first pattern vibroabsorptive coatings are applied to structures which border directly on the source of vibrations and its compartment. This pattern ensures attentuation of vibration on the path from the source to the compartment where reduction in vibration and noise is required. Such a pattern is preferable when it is necessary to reduce vibration and noise in many compartments, situated at some distance from the source of vibrations.

The second pattern envisions application of a vibroabsorptive coating on enclosures of a compartment with the goal of reducing their vibrations and noise they radiate into the compartment. Such a pattern is convenient in those cases when a reduction in vibration and noise is required in a small number of compartments.

In the third pattern coatings or other means of vibration absorption are applied to a single structure to reduce its vibrations and noise that it radiates at resonant frequencies. An example of such a pattern is application of a vibroabsorptive coating on the hull of a ship in the area of the screw propellor or damping individual elements of ship structures which rattle under action of traveling vibrations of the hull-frame, or other sources.

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More detailed recommendations on the use of means of vibration absorption on ships are contained in works [8, 34, 38, 41, 49, 55, 119]. Specifically, these works point out that in use of the first pattern a vibroabsorptive coating is effective when applied on machinery foundations and the floor of the send bottom of the engine room where the machinery is installed. For reduction of air noise in the compartments located over the engine room it is advantageous to apply a coating to the ceiling, sides and bulkheads of the engine room. As regards noise in compartments located close to the bow or stern away from the engine room, the effect of this coating will be small.

A general requirement for a given pattern of placement of a vibroabsorptive coating must be the necessity of applying it to all vibration-conducting paths which link the machinery with the area where compartments requiring noise reduction are situated. For example, it is senseless to apply a coating to separate hull-frame structures linking the engine area with a noisy compartment, leaving at the same time other structures undamped which run parallel to the ones damped.

From this point of view it is useful to apply a vibroabsorptive coating to pipes which link the source of vibration (machinery) with enclosures of the engine room. The absence of coatings on these pipes acn substantially reduce the effect of damping the foundation and other frame structures adjoining the machinery.

Some effect can be realized by applying a vibroabsorptive coating to bulkheads of the machine room. The effect of this coating is attributable to the increase in sound insulation of the bulkheads near the frequency of coincidence due to the rise in losses.

To damp hollow structures (foundation frame members, pillars, etc.) it is useful to fill them with sand or other friable vibroabsorptive material. In doing so, the change in their resonant frequencies and the possible coincidence of these frequencies with frequencies of the exciting forces must be kept in mind. If it is necessary to damp individual structures in a relatively narrow frequency spectrum local vibration absorbers can be used.

In the design of elements of ship structures which are to be damped one should keep in mind the advisability of using shapes which facilitate simplification of means of vibration absorption and increase their effectiveness. For example, the thickness of plates, to which vibroabsorptive coatings are to be applied, should be reduced as much as possible. It is advisable to build frame structures from tubular elements which are easily filled with vibroabsorptive materials, and not from profiled parts for which damping is complex.

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In some cases (particularly when it is necessary to normalize the noise level in compartments of a completed ship) it may be useful to employ removable vibration absorption structures which attach mechanically. A coating, consisting of a rubber layer (preferably perforated) and a sheet to press it down, can be used to damp plates [47]. The structure is attached with bolts, placed in a checkerboard pattern. Such a structure consitutes a combination stiffened-pliable vibroabsorptive coating and is effective over a wide spectrum of frequencies (Fig. 71). A removable installation for damping of pipes is described in work [88]. It consists of a rubber collar pressed against the pipe by a metal clamp (Fig. 72).

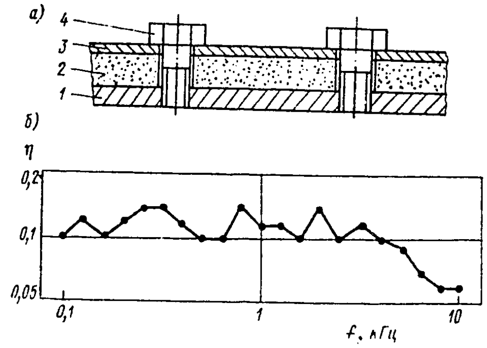

Fig. 71. Removable installation for damping of a plate (a) and loss factor of a 0.6 cm steel plate damped by this installation (b).

Key: 1. damped plate; 2. rubber viscoelastic layer; 3. pressing (stiffening) sheet; 4. mechanical fastening element.

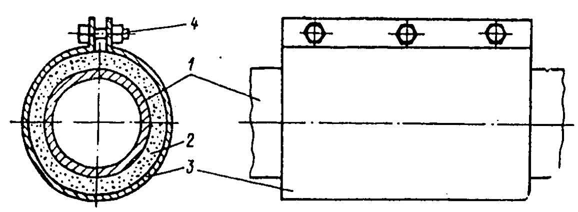

Fig. 72. Removable installation for damping of pipes [88].

Key: 1. damped pipe; 2. rubber viscoelastic layer; 3. metal sheath; 4. metal fastening element.

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V.S. Konevalov and V.V. Moiseyev suggest a vibration absorbing structure for damping ribs, which consists of a rubber gasket and a stiffening element attached with bolts. A schematic of this structure and the dependence of η on f are shown in Fig. 73.

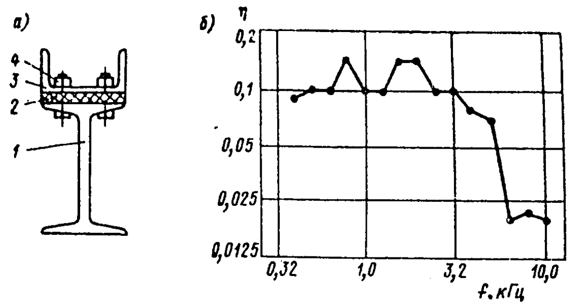

Fig. 73. Removable installation for damping of ribs (a) and loss factor of a rib damped with this installation (b).

Key: 1. damped rib; 2. rubber viscoelastic layer; 3. channel stiffening element; 4. mechanical fastening element.

Rattling of elements of ship structures can be eliminated by applying a vibroabsorptive coating to them, by using local vibration absorbers or by manufacturing them from vibroabsorptive laminated materials. The last method is recommended for use on ships also for manufacture of light-duty bulkheads, sound-insulating housings, ventilation ductwork, sheathing for cabin and engineroom walls, [payoles[1], etc.

Some effect (4-6 db at high frequencies) can be realized by applying vibroabsorptive coatings to machinery chassis or manufacturing them from vibroabsorptive alloys and materials.

The possibilities of using means of vibration absorption on other types of transport and in industry are shown in works [1, 39, 58, 60, 98, 101, 102, 103, 104, 106, 109, 115, 119].

§ 32. Examples of the Use of Means of Vibration Absorption on Ships

Technical literature, both at home and abroad, contains a large number of descriptions of cases where means of vibration absorption (mainly vibroabsorptive coatings) have been used on ships. An overview of these descriptions, accompanied by the necessary commentary, is presented below. This overview may prove useful in the design of patterns for application of vibroabsorptive coatings on ship structures.

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One of the first references to the use of vibroabsorptive coatings on ships is made in work [20]. On the motorship "Grünter" (FRG) a rigid vibroabsorptive coating 6 mm thick of sprayable mastic "Shalshuk" was applied to all enclosing structures of the engine room, except the bottom, for a total area of 200 m^2. The same coating was applied to sectors 3 m^2 each around the screw shaft bearings. A sector of the bottom over the propellor was covered with a layer of bitumen-cork mixture 25 cm thick. A schematic of this coating application is shown in Fig. 74. As one can see from the schematic, the vibroabsorptive coating was applied around the primary sources of vibration, which on this ship were the main engine, the drive line bearings and the propellors. A shortcoming of this pattern is the absence of a vibroabsorptive coating on the foundation of the main engine and the bottom of the engine room, i.e. in direct proximity to the primary sources of vibration where coatings are most effective.

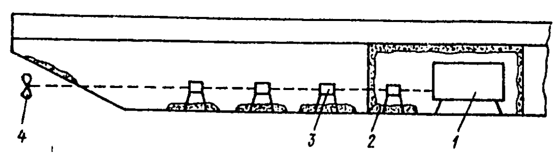

Fig. 74. Pattern of application of vibroabsorptive coating on the motorship "Grünten".

Key: 1. main engine; 2. vibroabsorptive coating; 3. propellor driveshaft bearings; 4. screw propellor.

On a "Raketa" motorship [8] they used a rigid vibroabsorptive coating of "Agat" plastic 2.5 mm thick with an intermediate layer of PCV-1 15 mm thick. This coating was applied to the enclosures and the deck in the wheelhouse, which is situated in part over the engine room, and on the bulkhead which separates the MO from the passenger salon. The total area coated amounted to 30 m^2 and the mass of the coating was 110 kg. According to the data of B.D. Tartakovskiy and V.B. Chernyshev, the noise level in the frequency spectrum 6.3-8.0 khz was reduced by 2-5 db in the wheelhouse and 1-6 db in the passenger salon. This effect is primarily attributable to a decrease in vibration of the enclosures bounding the engine room, where the main sources of vibration are located and, therefore, the highest levels of vibration exist.

On the floating crane "Bogatyr" [23] the walls of the living compartments and the deck, on which the compartments are located, were covered with a rigid vibroabsorptive coating of "Agat" plastic. The deck in the central control station was damped with "Neva-3U"

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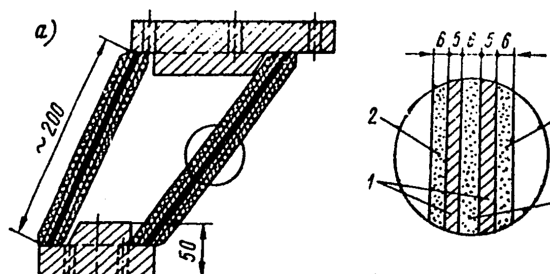
mastic. According to calcualted data, which were confirmed by experiment, the effect realized from these measures reached 7-11 db in the cabins and 2-5 db in the control station. In this case the designers employed the second pattern for equipping the ship with vibroabsorptive coating (see $ 31), in which the coatings are applied directly to the vibrating enclosures of the compartments in which a reduction in air noise level is required.

The hydrofoil passenger craft "Voskhod-2" was equipped with a rigid vibroabsorptive coating of "Agat" plastic with an intermediate layer of PCV-1. The coating was applied to the bottom, bulkheads and longitudinal screens, which enclose the reducing gear, as well as to the deck of the wheelhouse. Such a pattern of application of vibroabsorptive coatings corresponds to the first standard pattern examined in $ 31 and entails damping of structures situated in direct proximity to the source of vibration, which in this case is the reducing gear of the main engine.

Work [38] refers to the use of vibroabsorptive materials (Neva-3U mastic and bitumen) on river dry-cargo vessels with displacement of 1,000 tons and on a tug with a displacement of 300 tons. The effect of using these materials in combination with other means of vibration absorption amounts to 10-20 db.



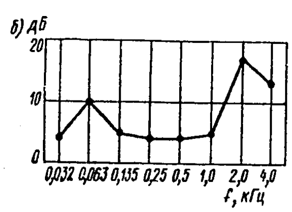


Fig. 75. Structure of a damped foundation and drop in level of vibration in this foundation.

Key: 1. steel plates; 2. vibroabsorptive coating; 3. asbestos.

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On the dredge "Francuis" (FRG) they employed a complex system of measures to combat air noise in the compartments, including a significant number of means of vibration absorption [116]. Fig. 75,a, depicts the structure of the intermediate foundation of an internal combustion engine. It can be seen that the walls of this structure consist of two sheets 5 mm thick separated by a layer of asbestos to prevent the metal sheets from touching and to increase the absorption of vibration in the structure. Using a double wall instead of a single wall for the foundation allowed its thickness to be reduced and and the same time increased attenuation within them of sonic vibrations. In addition, joining the thin plate to the massibe mounting slabs of the foundation offers increased vibration insulation which increases the actual loss factor of the damped sheets of the foundation (see $ 26). The external surfaces of the sheets of the foundation are damped by spraying with a rigid vibroabsorptive coating 6 mm thick. Fig. 75,b, shows the frequency characteristics of the reduction in vibration levels in the foundation of the structure described. This reduction ranges 4-17 db in the 0.032-4.0 khz frequency spectrum.

A vibroabsorptive coating was also applied to the foundations of some mechanisms, the floor of the second bottom and side walls of the engine room, which resulted in a reduction of air noise level by 6-8 db. In addition, vibroabsorptive material was introduced into the hollow rim of the idle pinion of the reduction gear and the wall of the forward rudder channel was damped with the vibroabsorptive coating "Remafon," consisting of a polyvinylchloride plastic with mica additives. This measure reduced the air noise level in the forward helm compartment by 4 db.

The scope of use of vibroabsorptive coatings is often very significant. On the floating hotel "France" (FRG) a vibroabsorptive coating was applied to the bottom in the area of the engine room and beyond it all the way to the first passenger cabins [95]. On the ferry "Princess Paola" (Belgium), which has a displacement of 3,600 tons, all walls of the engine room, for a total area of 2,000 m^2, were covered with a vibroabsorptive coating produced by the "Shell" firm. On the tug "Serklas" (Belgium), which has a displacement of 130 tons, the weight of the vibroabsorptive coatings used is about one ton. A coating of the type V.110 was applied on this ship to walls of the engine room, the wheelhouse and the cabins, as well as to the engine foundations on both sides of the deck [90].

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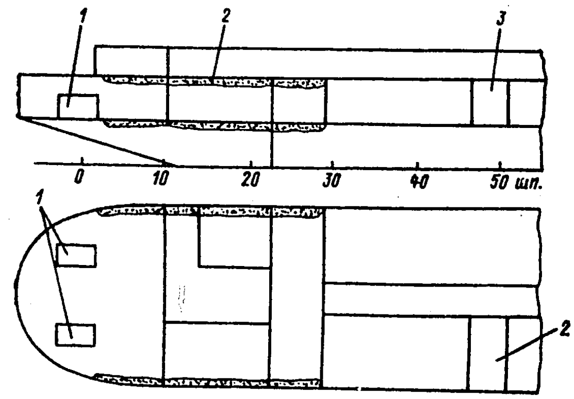

Fig. 76. Pattern of application of vibroabsorptive coating on the motorship "Rotterdam".

Key: 1. auxiliary diesel generator; 2. vibroabsorptive coating; 3. test cabin.

On the passenger ship "Rotterdam" (FRG) they used a rigid sprayable vibroabsorptive coating "Shalshuk 2K601" up to 10 cm thick [63]. The coating was applied on two decks and the side walls between the auxiliary diesel generator compartment and the passenger cabins over an expanse of 25 spaces (Fig. 76). The total area coated is about 400 m^2. The thickness of the damped structure is 0.5 cm. The effect of using the described damping scheme, relative to vibrations of the referenced diesel generators, was determined by change in the noise level in a cabin located 10 m forward of the boundary of the coated part of the ship. It should be noted that between the point of excitation of the ship's hull-frame and the test cabin there are undamped paths for transmission of vibratory energy. They include the upper deck and the keel of the ship as well as internal longitudinal bulkheads.

The frequency characteristics of the effect of using the coating is shown in Fig. 77. It can be seen that the effect is insignificant and does not exceed 5 db. It will be noted that the coating was applied before the ship was outfitted with equipment. It can be maintained that the influence of the undamped paths of transmission of vibratory energy, noted above, did not allow the effect to manifest itself in full measure. For comparison, Fig. 77 shows the frequency characteristics of the reduction in air noise level in the test cabin due to saturation of the ship with various equipment and trimming of the structures with decorative materials. Damping of all paths of transmission of vibratory energy, with all the noted factors in force,

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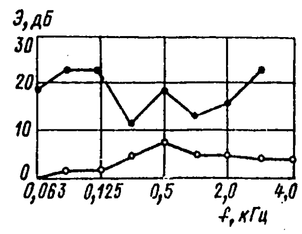

Fig. 77. Reduction of noise levels in a test cabin of the motorship "Rotterdam".

Key: O - after application of vibroabsorptive coating;
● - after saturation of the ship with equipment and decorative trim materials

led to a reduction in the noise level in the cabin of 20 db. From this the conclusion can be drawn that a more rational placement of vibroabsorptive coatings would have had a significant effect. In connection with this one cannot agree with the author of work [63] on the lack of promise in using vibroabsorptive coatings on ships for the purpose of reducing noise levels in compartments.

There is also information available on the localized use of vibro-absorptive coatings on ships. Work [96] speaks of damping walls of sound-insulating housings for noise-generating machinery. Work [72] describes a two-stage shock-absorbing mount for mechanisms in which the intermediate frame is faced with a vibroabsorptive coating. Some examples of the use of means of vibration absorption are given in work [34].

CONCLUSION

Modern means of vibration absorption (vibroabsorptive coatings and structural materials, localized vibration absorbtion, etc) represent an effective means of combating vibrations and air noise on ships. With the rational use of means of vibration absorption, noise levels in ship compartments, attributable to operation of machinery and other sources of vibration, can be reduced by 10-20 db. At the same time poorly conceived use of these means may not realize the required effect and result in an unjustified increase in displacement and higher ship cost.

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Developed methods permit evaluation of anticipated effectiveness of means of vibration absorption relative to vibrations and ir noise in compartments during the design phase. The most reliable and graphic method of such evaluation is electrical modeling of the ship's hull-frame using special analog installations or electronic computers. By using this method one can, without resorting to complex and expensive full-scale experiments, develop various patterns of use of means of vibration absorption on a specific ship and select from them the most rational from the point of view of effect and cost.

The use of means of vibration absorption must, as a rule, be provided for in the process of designing a ship, since only in this case is it possible to achieve greatest effectiveness of these means with minimum volume of materials used. At the same time, means of vibration absorption are one of the few soundproofiing methods which can be used on a completed ship in case it is necessary to further reduce the levels of vibration and noise in ship compartments.

Means of vibrations absorption aid in extending the life of elements of ship structures and equipment, which are prone to intense vibration and fatigue damage attributable to this vibration. Means of vibration absorption can be useful for reduction of vibration and noise which stem from machining of structural elements in shipbuilding yards and other metalworking plants.

Further studies in the area of development and perfection of means of vibration absorption and their use on ships should be conducted in the following directions:

-- creation of more effective, inexpensive vibroabsorptive coatings and construction materials suitable for practical use;

-- accumulation and generalization of experience in the use of means of vibration absorption on ships for the purpose of developing additional recommendations for the effective and rational practical application of these means;

-- improvement of methods of predicting the effectiveness of means of vibration absorption, relative to vibration and air noise in compartments, during a ship's design phase;

-- studies of the factors which limit the effectiveness of using means of vibration absorption (oscillation of the non-flexural type in damped structures, transmission of vibratory energy along rigidity ribs, etc.) and development of recommendations and means to reduce the influence of these factors.

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The results set forth in this book can be used in other spheres of industry where there is a need to reduce vibrations and noise in means of transportation or in enterprises.

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